W.K. McFarling

440

McFarling 1604 Polk

boiler hp 119

enthalpy
entropy 29

steam rate

STEAM, AIR, AND GAS POWER

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STEAM, AIR, AND GAS POWER

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SECOND EDITION

REWRITTEN AND RESET

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HOWARD E. DEGLER

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PREFACE TO THE SECOND EDITION

When the first edition of this textbook was published the Steam Tables of the American Society of Mechanical Engineers were not available. These newer data relative to the Properties of Steam were compiled by Professor J. H. Keenan and are included in this second edition in abridged form for both saturated and superheated steam.

Developments in the practice of power generation have made it necessary to remove from the book such material as was considered obsolete and of least importance. These sections have been replaced with descriptions and illustrations of recent power-plant equipment. The arrangement of the subject matter has been changed in a number of places. In the chapter Valves, Valve Gears, and Governors for Slide-valve Steam Engines a short discussion of valve diagrams has been introduced. Entirely new and a larger number of problems for classroom work have replaced those of the first edition.

The scientific and engineering symbols and abbreviations used in this book are in general accordance, except for the use of a period after each abbreviation, with the American Tentative Standards approved by the American Standards Association, of which the American Society of Mechanical Engineers is a member.

The authors wish to thank Professor W. N. Espy and Mr. R. F. Larson, of the University of Illinois, also Professor B. E. Short and Mr. M. M. Heller, of the University of Texas, for suggestions made relative to the revision of the book.

W. H. S.
H. E. D.

Urbana, Illinois,
April, 1933.

PREFACE TO THE FIRST EDITION

This book has been prepared as an elementary text for use in courses dealing with Heat Engineering, where only a limited amount of work in the subject can be included. The work is intended for students taking only a single course, for those registered in an accompanying Mechanical Engineering laboratory course, or for those who will register in an experimental laboratory course later in the curriculum.

The aim has been to describe briefly and clearly typical and representative equipment, and to explain the theory of such machines and devices. Due to the great number of products in the market, only a few illustrating the points under consideration could be included. The mathematical calculations involved are of the simplest order and illustrative problems have been included wherever necessary. The subject matter dealing with Steam Engines has been reduced to a minimum; much material formerly included in similar textbooks is omitted.

The authors express their thanks to Professor G. A. Goodenough for permission to include data from his work, "Properties of Steam and Ammonia," and for aid in preparing certain portions of the text. The authors also thank Professors A. C. Willard, A. P. Kratz, and J. A. Polson, and Dr. John A. Goff for assistance rendered in reading and commenting on the text material. Appreciation of the cooperation received from the various manufacturers, whose equipment is illustrated, is expressed and wherever possible the use of such illustrations has been acknowledged.

W. H. S.
H. E. D.

Urbana, Illinois

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STEAM, AIR, AND GAS POWER

CHAPTER I

INTRODUCTION

1. Heat.—When two bodies at different temperatures are brought together the temperature of the hot body falls, that of the cold body rises, and the change continues until the two bodies are at the same temperature. The decrease of temperature of one body is due to the loss of "heat," and the increase in temperature of the other is due to the "heat" received by it.

Heat may be defined as *a form of molecular energy*, the movement of the molecules increasing as the body is heated and decreasing as it is cooled. The higher temperature of a hot body indicates that the movement of the molecules is greater than that in a cold body. When these two bodies are brought together the molecules of the hot body impart motion to the molecules of the cold body; thus the molecular activity of the hot body decreases and that in the cold body increases until equilibrium is established. The amount of heat, above any reference point, that a body can transmit or absorb depends upon its weight, its unit thermal capacity (specific heat), and its temperature.

2. Temperature.—*Temperature* is a measure of the intensity of heat as distinguished from the quantity of heat; it is *the thermal state of a body considered with reference to its ability to transmit heat to other bodies.*

The scale of the constant-volume hydrogen thermometer is taken as the standard temperature scale. Within the limits of ordinary use the scale of the mercury thermometer agrees closely with the standard scale, but above 600 deg. F. the divergence between the two scales may be appreciable.

The space within the mercury thermometer tube, Fig. 1, not filled by the mercury is usually a vacuum in thermometers for measuring ordinary temperatures, up to 600 deg. F. For higher temperatures, up to 1000 deg. F., and for greater accuracy at ordinary temperatures the space is filled with an inert gas, nitrogen or carbon dioxide under pressure, to prevent vaporization of the mercury. The low temperature limit for the mercury thermometer is −39 deg. F. For lower tem-

peratures, alcohol, pentane, or petroleum ether may be used as the thermometric substance.

The position of the top of the mercury column when the lower part of the thermometer is immersed in freezing water is called the *freezing point*; this is marked on the glass tube itself or on a scale attached to the tube. The position of the top of the mercury when the lower part of the thermometer is immersed in boiling water at "standard atmospheric pressure" (14.696 lb. per sq. in.) is called the *boiling point*; this is marked the same way as the freezing point. The range between the freezing point and the boiling point is called the *fundamental interval* of the temperature scale.

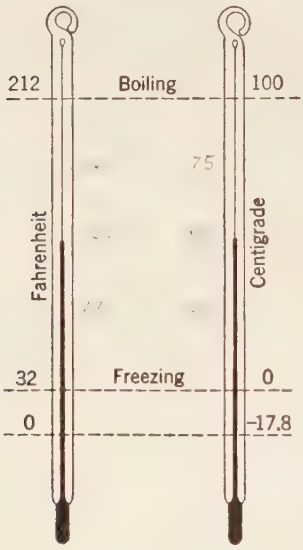


FIG. 1. —Thermometer scales.

On the Centigrade scale the fundamental interval is divided into 100 equal parts; the freezing point is taken as zero and the boiling point as 100 deg. On the Fahrenheit scale the fundamental interval is divided into 180 equal parts; the freezing point is taken as 32 deg. and the boiling point as 212 deg. The relation between the two temperature scales is shown diagrammatically in Fig. 1. If t_c denotes a temperature on the Centigrade scale, and t_f denotes the same temperature on the Fahrenheit scale, then

$$t_f = \tfrac{9}{5}t_c + 32, \tag{1}$$

and

$$t_c = \tfrac{5}{9}(t_f - 32). \tag{2}$$

Pyrometers [1] are generally used for measuring temperatures exceeding 800 deg. F. Metallic or *mechanical pyrometers* are made of two metals having different rates of expansion, such as iron and brass, or graphite and iron. The metallic rods are fused together at one end (hot junction); the other or free ends are connected by gears and levers to move a pointer over a dial graduated in degrees of temperature. Although the use of such pyrometers is quite common, they are generally unreliable and should not be used for temperatures above 1000 deg. F.

The *electrical-resistance pyrometer* depends upon the increase of electrical resistance of certain metals with the temperature. Platinum

[1] For full information concerning pyrometers see "Pyrometric Practice," by Foote, Fairchild, and Harrison; U. S. Bureau of Standards Technologic Paper 170, 1921.

is usually selected because for a given temperature it has a very constant resistance and it does not deteriorate at high temperatures. Platinum wire is wound upon a mica framework and the variation of resistance is measured by means of a small Wheatstone bridge; the adjustable resistance of the bridge may be graduated to indicate the temperature in degrees. Up to 1000 deg. F. the error should be less than $\frac{1}{50}$ deg. F., and at 2400 deg. F. not more than $\frac{1}{2}$ deg. F.

Thermoelectric pyrometers measure temperatures by means of the electric current generated in a "thermocouple" made by fusing together the ends of two wires made of different metals. The voltage of the thermocouple increases with the temperature to which one fused end of the thermocouple is subjected, the temperature of the opposite end being kept constant. A sensitive galvanometer or voltmeter, usually graduated in degrees of temperature, measures the voltage. A typical thermoelectric pyrometer installation is shown by Fig. 2. A couple made of one wire of pure platinum with another of 90 per cent platinum and 10 per cent rhodium is not injured at 3000 deg. F., although satisfactory thermocouples are also made of nickel steel and copper, nickel-chromium, and nickel-aluminum alloys.

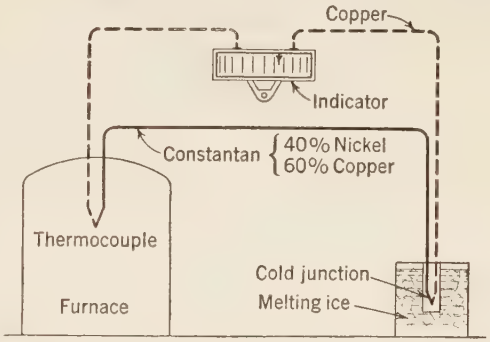


FIG. 2.—Typical thermoelectric pyrometer installation.

The temperature of a body may be obtained from a measurement of the intensity of the radiant energy it emits. In a *radiation pyrometer* the intensity of all wave lengths is measured. Usually the energy is focused upon the hot junction of a small thermocouple, and the voltage generated is measured by a potentiometer or by a galvanometer of high resistance. Such instruments are particularly serviceable for determining the temperature of a bed of fire, as in a boiler furnace. An *optical pyrometer* measures the intensity of a narrow spectral band of radiation emitted by a glowing object. It is simply a photometer using monochromatic light (usually red) in which the intensity of radiation from either a constant or standard source (electric lamp, oil flame, etc.) is compared with that from the object of which the temperature is desired. The two intensities may be made equal by adjusting various types of absorbing devices (absorption glasses, iris diaphragms, etc.) on the furnace side or on the standard lamp side, depending upon

which side is the brighter. When an optical pyrometer is sighted on bodies in the open it may have an error of 10 per cent or more, whereas the reading with a radiation pyrometer may be 40 per cent too low.

Pyrometric cones, also called Seger cones, are pyramids about 2 in. high made of various mixtures of kaolin, feldspar, lime, magnesia, quartz, aluminum oxide, iron oxide, lead oxide, and boric acid to give a definite melting point. The melting points of the various cones range from 1000 to 3500 deg. F. by steps of 50 to 100 deg. F., each cone having a definite number. The temperature corresponding to a given number has been reached when the tip of the cone turns over and touches the surface on which its base is resting. The rate of heating and the character of the surrounding atmosphere (oxidizing or reducing) affect the melting temperatures.

A great number of recording thermometers operate by the expansion of the vapor of ether, mercury, or other liquids confined in a bulb and capillary tube connected to a pressure-measuring device. When the bulb is heated, the vapor tension increases and actuates the pressure-recording device.

3. Absolute Temperature.—Experiments show that the amount by which air, hydrogen, or any other so-called perfect gas expands per degree Fahrenheit of increase in temperature—as in passing from 32 to 33 deg. F. at atmospheric pressure—is about $\frac{1}{491.6}$ of the original volume. When the scale of the gas thermometer is taken as the standard it may be shown that, when 491.6 cu. in. of gas at 32 deg. F. are heated under constant pressure to 33 deg., the volume will be 492.6 cu. in. Similarly, if the gas is cooled from 32 to 31 deg. F. its volume changes from the original 491.6 to 490.6 cu. in. If the same law could be held to apply down to the lowest temperatures, the volume will become zero at 491.6 degrees below 32 deg. F., or at a temperature of -459.6 deg. F. (practically -460 deg. F.).

If the temperature is reckoned not from the freezing point but from a zero which is 491.6 Fahrenheit degrees below freezing, the volume of a gas when heated under constant pressure is proportional to the temperature reckoned from that zero, called the *absolute zero of temperature.* This absolute zero is sometimes defined as the point which marks the absence of heat energy or molecular motion in a body.

Denoting the temperature on the ordinary scale by t and the corresponding temperature reckoned from the absolute zero by T,

$$\text{on the Fahrenheit scale, } T_f = t_f + 460, \qquad (3)$$

$$\text{on the Centigrade scale, } T_c = t_c + 273. \qquad (4)$$

The "thermodynamic" scale of temperature starts from a zero which is absolute in the sense that no lower temperature can exist; the zero of the thermodynamic scale coincides with the absolute zero of the gas scale.

4. Transmission [2] and Measurement of Heat.—There are three methods by which heat may be transmitted or conveyed from one place to another, namely, *conduction*, *convection*, and *radiation*. Heat passing from one part of a body to another part of the same body, or from one body to another in physical contact with it, without displacement of the particles of the body, is said to flow by conduction. Convection is the transfer of heat from one place to another within a fluid (gas or liquid) by the mixing of one portion of the fluid with another. When air comes in contact with a hot body the air is heated and rises; heat carried away in this manner is transferred by convection.

All bodies give off heat in the form of radiant energy, which is disseminated in all directions as a wave motion in the ether, similar to the light that is emitted from a lamp. Radiation falling upon a body is absorbed by it either wholly or in part. When two bodies, one at a higher temperature than the other, are placed within an enclosure, there is an interchange of energy between them. The hotter body radiates more energy than it absorbs, and the colder body absorbs more than it radiates. After equilibrium of temperature is established the process continues, each body radiating and absorbing energy.

The quantity of heat is measured in terms of the *British thermal unit* (*B.t.u.*), which is defined by the American Society of Mechanical Engineers as $\frac{1}{180}$ of the heat required to raise the temperature of one pound of water from 32 to 212 deg. F. Thus the British thermal unit is the average amount of heat required to raise the temperature of a pound of water through one degree Fahrenheit, between the conditions of freezing and boiling at atmospheric pressure (14.696 lb. per sq. in. abs.).

5. Specific Heat.—The *specific heat* (sometimes called unit thermal capacity) of any substance is the amount of heat required to raise the temperature of a unit weight of the substance one degree. The specific heat of all substances varies with the temperature, and since all substances vary in volume or pressure with temperature changes, it is necessary to distinguish between specific heats at constant volume and at constant pressure, expressed ordinarily as c_v and c_p, respectively.

Because of the low coefficients of expansion for liquids and solids,

[2] For additional information concerning the transmission of heat see "The Basic Laws and Data of Heat Transmission," by W. J. King; *Mechanical Engineering*, vol. 54 (1932), pp. 190, 275, 347, 410, 492, and 560. Also "Heat Transmission," by William H. McAdams; McGraw-Hill Book Company, Inc., 1933.

there is but little difference in the specific heat at constant volume and that at constant pressure. With gases, on the other hand, the difference is appreciable. When heat is added to a gaseous substance its volume may be kept constant, in which case no external work is done; or the gas may be allowed to expand during the addition of heat, the pressure being kept constant. The specific heat at constant volume, therefore, is always less than that at constant pressure by an amount equal to the heat required to do the work of expansion against external pressure.

In both specific heats, at constant volume and at constant pressure, it is necessary to distinguish between *instantaneous* and *mean* specific heat. The instantaneous specific heat of a substance is the amount of heat that must be added to a unit weight of such material at some definite temperature to increase its temperature one degree under given conditions of volume and pressure. The mean specific heat of a substance, over a given temperature range, is the average quantity of heat required to raise the temperature one degree of a unit weight of the substance under the conditions of volume or pressure which exist. In practically all boiler and combustion work, the mean specific heats are used.

If a pound of water is considered as a standard, the mean specific heat over a temperature range of 32 to 212 deg. F. from the definition of a B.t.u. is unity. The specific heats of air at constant volume (c_v) and at constant pressure (c_p) vary with the temperature. A number of determinations of these values have been made, and generally accepted values for the mean specific heats are:

c_v for air, 32 to 400 deg. F..... 0.173,
c_p for air, 32 to 400 deg. F..... 0.242,
Ratio of the specific heats $\frac{c_p}{c_v}$..... 1.40.

6. Relation of Energy, Work, and Power with Heat.—Energy is the capacity for doing work and the ability to overcome resistance. Energy exists in many forms, as heat, mechanical, electrical, chemical, etc. Several units are used in measuring energy, and each type of energy has its characteristic unit of measurement; for heat energy the unit is the B.t.u., and for mechanical energy the *foot-pound*. The latter is defined as the capacity for lifting a one-pound weight through a vertical distance of one foot.

Potential energy is energy due to the relative position of one body to another, or of one part of a body to another part, when the two bodies or parts of the same body are under the action of a force or

forces tending to alter their relative positions. For example, a body which is allowed to fall toward the earth may be made to do work; hence, before it begins to fall it possesses potential energy, because if allowed to resume its unrestrained form the body can be made to do work. Likewise, compressed air possesses potential energy. The energy stored in a piece of coal is potential energy, and under favorable conditions the atoms of the constituents of the coal and the atoms of the oxygen of the air will rush together and produce heat, which may be converted into work.

Kinetic energy is energy due to the motion of a body. Ten pounds of water at rest at a height of 100 ft. above sea level possess 1000 ft-lb. of potential energy. If this water is allowed to fall freely to sea level, without doing work on the way and without friction or other losses, it will in any position of its fall possess 1000 ft-lb. of energy; as it descends, the potential energy will decrease, and the remainder of the 1000 ft-lb. will be stored in the water as kinetic energy. When the water has fallen 25 ft. its potential energy will be 750 ft-lb. and its kinetic energy will then be 250 ft-lb.

When a force acting on a body causes that body to move, the force is said to do *work*. In general, work is the overcoming of resistance through a distance. In measuring mechanical work the unit which is used by engineers is the work done when a force of one pound acts through a distance of one foot; this unit is the foot-pound. Thus, if F represents the magnitude of the force and l the distance through which it acts, the work

$$W = Fl. \tag{5}$$

If F is in pounds and l is in feet, the work will be expressed in foot-pounds.

Power is the rate at which work is performed, or the number of units of work performed in unit time. A heat engine or any working agent is said to be developing one *horsepower* when it does 33,000 ft-lb. of work in one minute. Using the same notation as for equation 5 and representing by t the time in minutes required to do the work,

$$hp = \frac{Fl}{33{,}000t}. \tag{6}$$

The unit of electrical power is the *watt*, which is the power of an electrical current of one ampere flowing under an electromotive force (electrical pressure) of one volt. Thus, volts times amperes equals watts. A horsepower is equal to 746 watts, and one kilowatt equals 1000 watts or 1.34 horsepower.

That heat may be produced from mechanical work is a matter of common knowledge. Work is consumed in overcoming the friction between the contact surfaces of the moving parts of machines, showing that some of the work of driving the machine has been transformed into the heat required to create the rise in temperature.

On the other hand, work may be obtained by the expenditure of heat, as illustrated in the steam engine, steam turbine, gas engine, and other heat engines. In the steam engine the steam in expanding partly condenses and also has its temperature decreased. The heat energy thus given up is transformed into the work obtained from the engine. Recent experiments [3] made at the U. S. Bureau of Standards show that 778.57 ft-lb. of work are expended in producing one B.t.u. of heat. This relation between work and heat is called the *mechanical equivalent of heat;* the figure 778 is generally used in engineering calculations.

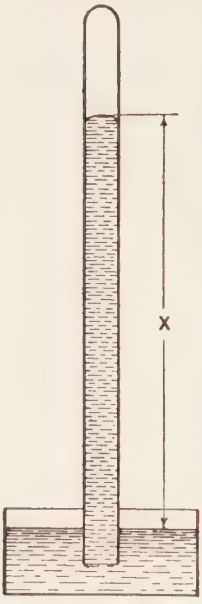


Fig. 3.— Mercurial barometer.

7. Pressure.—When pressure is measured in atmospheres it is measured in terms of a unit which approximates the vertical air pressure on a horizontal unit area at the earth's surface. This unit then represents the weight of the superincumbent atmosphere, it being assumed that the atmosphere is at rest horizontally and vertically. In English units this standard atmospheric pressure amounts to 14.696 lb. per sq. in.

Barometers are instruments used to measure the pressure of the atmosphere, such pressure being commonly called the barometric or atmospheric pressure. There are two principal types, the mercurial barometer and the aneroid barometer. The mercurial barometer is the one generally used and recommended for accuracy. Figure 3 is a diagrammatic illustration of the Fortin type, as used by the U. S. Weather Bureau. Essentially, the instrument consists of a glass tube, about $\frac{1}{4}$ in. in internal diameter and about 3 ft. long, closed at one end and filled with mercury; the open end of the tube is then inverted into a small cistern of mercury, as shown in Fig. 3. Above the mercury in the glass tube is a high vacuum, giving essentially zero pressure on top of the column. The atmospheric pressure depresses the mercury in the cistern, so that the column of mercury in the glass tube measured above the level in the cistern exactly balances the atmospheric pressure. The

[3] "Report on Progress in Steam Research at the U. S. Bureau of Standards," by Osborne, Stimson, and Fiock; *Mechanical Engineering*, vol. 50 (1928), p. 152.

exact height X of the top of the mercury column in Fig. 3 is read by means of a scale fastened to a brass shield which surrounds the glass tube. A sliding vernier is provided so that precise readings may be obtained. A thermometer is always attached to indicate the temperature of the column.

The atmosphere exerts a variable pressure, depending upon the altitude and the weather conditions. Observation has shown that the average atmospheric pressures at sea level and 45 degrees latitude is approximately 14.7 lb. per sq. in. By international agreement this unit of *standard atmospheric pressure* has accordingly been adopted as

> 760 mm. of mercury (at 32 deg. F.) = 29.921 in. under the international standard gravity of 32.174 ft. per sec. per sec.

Hence, in reducing volume measurements of gases and vapors to standard conditions of pressure, the reading of the barometer must be corrected for expansion, and the height of the mercury column must be reduced to what it would be at 32 deg. F.

What is directly measured in the barometer is the length of a column of mercury; the weight of this column gives the pressure per unit area of cross-section. A cubic inch of mercury at 32 deg. F. weighs 0.491 lb., and for ordinary temperatures this weight may be used with sufficient accuracy, or a column of mercury one inch high exerts a pressure of 0.491 lb. per sq. in. Hence, a reading in inches of mercury is changed to pounds per square inch by multiplying the height of the mercury column by 0.491.

Gages indicating pressures above that of the atmosphere are called pressure gages, and those showing pressures less than atmospheric are known as vacuum gages. Pressures are usually measured with Bourdon gages (see Fig. 46) consisting of an elastic metal tube of oval cross-section, bent into an arc. Gages of the tube type may be used for determining the pressure of steam, water, air, ammonia, etc. The gage pressure plus the atmospheric pressure, both expressed in the same units, is called the *absolute pressure*, i.e., absolute pressure is the pressure above zero pressure. For cases where a measured pressure is below atmospheric, the pressure indicated by the corrected reading of a U-tube (Fig. 129) or a vacuum gage is called *vacuum*, usually measured in inches of mercury, and may be expressed with a minus sign. The absolute pressure in such cases is the difference between the barometric reading and the vacuum gage reading, or the algebraic sum of the two readings.

For greater accuracy at low pressures, U-tubes (Fig. 129) filled with mercury, water, kerosene, or other liquids are often used. The difference in level of the liquid in the two legs, expressed in inches of mercury

or water, is the most common measure of the pressure (or vacuum). When water accumulates on top of a mercury column, the length of the mercury column must be corrected. This correction is most conveniently made by dividing the length of the water column by the specific gravity of mercury (13.6) and adding this equivalent length to the mercury column on which the water rests. An atmospheric pressure of 14.7 lb. per sq. in. is equivalent to a "head" of approximately 34 ft. of water at ordinary temperatures.

Example.—Find the absolute boiler pressure when the pressure gage shows 148 lb. per sq. in. and the barometer reads 29.45 in. of mercury.

Solution.—Atmospheric pressure = $29.45 \times 0.491 = 14.46$ lb. per sq. in.; hence, absolute boiler pressure = $148 + 14.46 = 162.46$ lb. per sq. in.

Example.—Find the absolute pressure in a condenser when the vacuum gage indicates 28.20 in. of mercury and the barometer reads 29.58 in. of mercury.

Solution.—Absolute condenser pressure = $29.58 - 28.20 = 1.38$ in. of mercury; or $1.38 \times 0.491 = 0.678$ lb. per sq. in.

PROBLEMS

1. A temperature rise of 40 deg. C. was permissible in the windings of an electric generator. Express the temperature rise in deg. F.

2. Cooling water entered a condenser at a temperature of 60 deg. F. and left at 78 deg. F. Find the temperature rise in deg. F. and deg. C.

3. Water entered a feedwater heater at a temperature of 80 deg. F. and left at 200 deg. F. (*a*) What was the temperature rise in deg. C.? (*b*) Find the exit temperature in deg. C. (*c*) Express the entering temperature in deg. F. absolute. (*d*) What was the exit temperature in deg. C. absolute?

4. How many B.t.u. were required to raise the temperature of 10 lb. of feedwater from 110 deg. F. to 200 deg. F.?

5. Find the B.t.u. required to heat one pound of each of the following substances from a temperature of 20 deg. F. to 145 deg. F.

(*a*) Ice, $c_p = 0.5$ latent heat of fusion = 144 B.t.u. per lb.

(*b*) Air at constant pressure.

(*c*) Air at constant volume.

(*d*) Iron, specific heat = 0.101.

(*e*) Aluminum, specific heat = 0.207.

(*f*) Copper, specific heat = 0.0915.

6. Three hundred thousand cubic feet of air were supplied to a building per hour for heating and ventilation purposes. If hot water was used to heat this air from 35 to 75 deg. F., find the gallons of water required per hour if the water temperature was decreased from 180 to 150 deg. F. Density of air, 0.074 lb. per cu. ft.; neglect heat loss from piping. Assume that the heat was added to the air at constant pressure.

7. When 1.2 grams of gasoline were burned in a calorimeter, the temperature of 2500 grams of water increased from 71.1 to 80.5 deg. F. If all the heat produced was absorbed by the water, compute the B.t.u. which the combustion of one pound of this gasoline liberated.

8. The piston of a hydraulic elevator was 10 in. in diameter, and it had a pressure of 90 lb. per sq. in. acting upon it. Find the work done during 2 ft. of travel.

9. The air pressure upon the front of a train was 30 lb. per sq. ft. when the train traveled 50 miles per hour. If the train presented an area of 75 sq. ft., find the horsepower required to overcome the wind resistance.

10. The piston of a steam engine traveled 600 ft. per minute, and the average force of the steam acting against the piston was 55,000 lb. Find the horsepower developed.

11. A waterfall has a height of 132 ft. Flow measurements indicated that 50,000 cu. ft. of water at 60 deg. F. pass over the fall each second. Calculate the potential horsepower.

12. (*a*) What is the heat equivalent of 389,000 ft-lb. of work?

(*b*) Calculate the B.t.u. equivalent of one horsepower-hour.

13. A weight of 10 lb. fell from a height of 3890 ft. into 60 lb. of water. If the energy were transformed into heat, find the rise in temperature of the water in deg. F.

14. A natural-gas engine delivered 1,980,000 ft-lb. of work at the crankshaft per minute.

(*a*) Find the horsepower delivered.

(*b*) If the engine operated in this way for 2 hours, how many horsepower-hours were made available?

(*c*) Calculate the heat equivalent of part *b* in B.t.u.

(*d*) Find the equivalent kilowatt-hours of part *b*.

15. A steam gage in a house-heating boiler indicated a pressure of 5 lb. per sq. in. and the barometer read 29.32 in. of mercury.

(*a*) Compute the absolute pressure in pounds per square inch.

(*b*) Express the gage pressure in inches of mercury.

(*c*) Change the gage pressure to feet of water.

16. (*a*) The total "head" against which a pump was operating was 326 ft. of water. Find the equivalent pressure in pounds per square inch.

(*b*) The draft gage connected to the furnace of a boiler read 0.85 in. of water. What pressure does this represent in pounds per square inch, if the temperature of the air surrounding the gage was 70 deg. F.?

17. If the gage on a steam boiler indicated a pressure of 195 lb. per sq. in., what was the absolute pressure in the boiler in pounds per square inch if the barometer reading was 29.67 in. of mercury?

18. The vacuum gage on a condenser read 27.61 in. of mercury and at the same time the barometer read 29.22 in. of mercury. What was the absolute pressure in the condenser in pounds per square inch?

19. A mercury U-tube manometer was used to measure the vacuum in a tank. The height of the mercury in one column of the manometer was 18.2 in. and in the other column was 8.7 in. Two inches of water rested on top of the mercury in the short column. Find the absolute pressure in the tank in pounds per square inch. Barometer 29.55 in. of mercury.

20. If, in problem 19, oil at a temperature of 90 deg. F. and a specific gravity of 0.90 had been used in place of the mercury and water, what would have been the difference in elevation of the two columns of the manometer in inches of oil?

CHAPTER II

PRINCIPLES OF THERMODYNAMICS

8. First Law of Thermodynamics.—The principle of the "Conservation of Energy" in relation to heat and mechanical work constitutes the First Law of Thermodynamics, which may be stated: *When heat is produced by the expenditure of mechanical work a definite quantity of heat appears for every unit of work done,* and conversely, *when heat is converted into mechanical work, a definite quantity of heat disappears for every unit of work done.*

When the brakes are applied to the wheels of a moving train, in order to bring it to rest, the kinetic energy of the train is converted into heat at the rubbing surfaces of the brakes and wheels; and if the wheels slide, heat is produced at the rubbing surfaces of the wheels and rails. The kinetic energy of the train will be transformed into a definite quantity of heat. Conversely, however, it cannot be said that a given amount of heat can be converted completely into work even in an ideal engine. The purpose of thermodynamics is to explain this statement and to determine the factors that influence the amount of heat that can be converted into work. A heat engine that converts the heat produced by the combustion of a fuel is an example of the conversion of heat to work.

Recent experiments made at the U. S. Bureau of Standards have shown that 778.57 (about 778) ft-lb. of work are spent in producing a unit of heat (B.t.u.); this quantity is called the *mechanical equivalent of heat* and is generally represented by the symbol J. The foot-pound is a unit derived from considerations of force and distance involved in the performance of mechanical work; the B.t.u., a unit derived from considerations of the change of temperature due to heating. That there is a constant relation between the two units (namely, one B.t.u. = 778 ft-lb.) simply means that mechanical work can be measured in B.t.u. as well as in foot-pounds, or that a quantity of heat can be measured in foot-pounds as well as in B.t.u. The symbol A is used for the reciprocal of J, i.e., $A = \frac{1}{J}$.

9. Second Law of Thermodynamics.—The first law states a relation that must be satisfied in any transformation of energy. The second

law gives information regarding the possibility of transformation of a given form of energy into mechanical work; this is called the availability of the energy. The Second Law of Thermodynamics has been stated in many ways, but essentially it is the law of the degradation of energy. Clausius' form of the second law is: *It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature.*

The second law states, in effect, that heat will not pass automatically from a colder to a hotter body. Heat can be forced to pass up, as in the action of a refrigerating machine, but only by applying an "external agency" to drive the machine. A heat engine acts by letting heat pass down from a hotter to a colder body, not by direct conduction one to the other but by making a medium alternately take in heat from the hot body and reject heat to the cold body. Thus the medium undergoes expansions and contractions in which its pressure is, on the whole, greater during expansion than during contraction; and in this way a *part* of the heat that is passing down through the engine is converted into mechanical work.

10. Working Medium.—The working medium is the vehicle by which heat passes through a heat engine. The mediums generally employed are fluids such as air, water, and water vapor (steam). Because these fluids have a capacity for taking in heat, they can act as vehicles for conveying heat from one temperature to another. In this process, volume changes occur; and by means of these changes on the part of the working medium, the heat engine does work. The fluid receives heat by being brought into contact with some source of heat at a comparatively high temperature; it transforms a small part of this heat into mechanical work, and rejects the remainder to what may be called a receiver of heat, the latter at a comparatively low temperature. The greater part of the heat may be said to simply pass through the engine from the source to the receiver, becoming, in the process, "degraded" as regards temperature.

The working medium of a heat engine usually returns periodically to the same state of temperature, pressure, volume, and physical condition. Each time this occurs, the medium is said to have passed through a complete *cycle* of operations. In such a complete cycle any mechanical work done must have been done at the expense of the heat taken in during the cycle, because the medium always returns to its original state and then has the same amount of internal energy (Art. 12) that it may have had at the beginning of the cycle. The application of the First Law of Thermodynamics for a complete cycle indicates

Heat taken in = Heat rejected + Work done by the medium.

11. Work Done.—In an engine of the usual cylinder and piston type the medium does mechanical work by changes of volume. Let the pressure of the working medium be P pounds per unit area while the piston moves a distance dl, this distance being taken infinitely small so that the pressure may be considered essentially constant during the displacement. If the area of the piston is a square feet, the total force on it is Pa, and the work done is $Padl$. But $adl = dV$, the change of volume, hence

$$dW = PdV. \tag{7}$$

A diagram may be drawn in which the relation of the pressure of any working medium to its volume is shown by rectangular coordinates, as in Fig. 4. The area $MABCN$ is obtained by the summation of elements of type PdV and is expressed by the integral, $\int_1^2 PdV$. But from equation 7 this gives the external work done by the medium; hence the external work done is represented by the area $MABCN$.

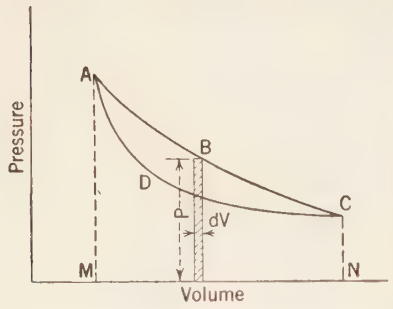


Fig. 4.—Pressure-volume diagram.

If the medium undergoes a complete cycle of change by expanding along curve ABC and by being compressed back through D to A, work is being done during expansion from A to C equal to $MABCN$; and work is spent during compression from C through D to A, equal to the area $MADCN$. The net amount of work done by the medium during the cycle is equal to the algebraic sum of these areas, thus equal to the area of the closed figure or diagram $ABCDA$. However, if the process were such as to trace the figure in the opposite direction, the medium being expanded along ADC and compressed from C to A through B, the enclosed area would be a measure of the work expended upon the medium.

12. The Energy Equation.—A medium which is being heated may expand, thus doing some mechanical work which in general will not equal the work equivalent of the heat added. If it is not equal, then the Principle of Conservation of Energy suggests that the difference must appear as an increase or decrease in the internal energy of the medium. Since there is no known method of determining the total internal energy of any medium, it is necessary to study changes in the internal energy as heat or work is added or removed from the medium. If heat were added to a medium and no work were done by

it, the internal energy of the medium would increase. When work is done on the medium without the addition of heat, the internal energy would likewise increase. Finally, if heat were added to and work were done by a medium,

Heat added to the medium

= Increase in internal energy + Work done by the medium,

or, if the quantities were small,

$$dQ = dU + dW.$$

In this equation dQ is considered positive if heat is absorbed by the medium and negative if heat is rejected by it. Also dW is positive when work is done by the medium and negative when work is done on it. In order to express all the terms in foot-pounds the equation is written as follows, since heat added is expressed generally in B.t.u. units, hence

$$JdQ = JdU + dW. \tag{8}$$

Substituting the value for dW given by equation 7 in the above equation,

$$JdQ = JdU + PdV. \tag{9}$$

At the end of a complete cycle the total internal energy U of the medium is the same as at the beginning, or the change in internal energy for the cycle is zero. In Art. 11 it was shown that the work done around a closed cycle is not in general equal to zero; thus the heat added is not equal to zero. The energy equation for a *closed cycle* may be written,

$$(Q) = (W).$$

The parentheses around the letters are used to represent the heat added around the cycle, i.e., the difference between the heat added and the heat rejected, or the net work done around the cycle.

13. Gas Equations.—To illustrate the application of the preceding general statements regarding transformations of energy to specific cases, it is convenient to consider the so-called perfect gases, because for these systems the relation between pertinent quantities (temperature, pressure, and volume) may be expressed simply, as in equation 14.

Boyle's law for a perfect gas is: *If the temperature is kept constant, the volume of a given weight of gas varies inversely as the pressure.* Thus,

$$Pv = C' \text{ (a constant)}, \tag{10}$$

or

$$P_1v_1 = P_2v_2 = P_3v_3 = \ldots . \tag{11}$$

One statement of Charles' law is: *When a perfect gas receives heat at constant volume the pressure varies directly as the absolute temperature,*

$$\frac{P}{T} = C'' \text{ (a constant)}, \tag{12}$$

or

$$\frac{P_1}{T_1} = \frac{P_2}{T_2} = \frac{P_3}{T_3} = \ldots . \tag{13}$$

A perfect gas is one that obeys exactly the laws of Boyle and Charles. Actually there is no perfect gas, but certain gases such as air, oxygen, hydrogen, nitrogen, and helium (permanent gases) very closely follow the above laws except at very low and very high temperatures.

Evidently the single relation that satisfies Boyle's and Charles' laws (compare equations 10 and 12) is

$$\frac{Pv}{T} = R \text{ (a constant)}, \tag{14}$$

where v is the volume of one pound of gas. Let M denote the weight of gas; then $V = Mv$, and substituting for v in (14),

$$PV = MRT, \tag{15}$$

or

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} = \ldots , \tag{16}$$

where P = absolute pressure in pounds per square foot.
V = volume of gas in cubic feet.
M = weight of gas in pounds.
R = gas constant (ft-lb., per lb., per degree).
T = absolute temperature, deg. F.

The value of the constant R for a given gas may be determined, if the specific volume is known for any given condition of pressure and temperature. The specific volumes of various gases at atmospheric pressure (14.7 lb. per sq. in. abs.) and 32 deg. F. are: air, 12.39; oxygen, 11.21; hydrogen, 177.9; and nitrogen, 12.80 cu. ft. per lb. Thus, for *air*,

$$R = \frac{14.7 \times 144 \times 12.39}{1 \times (459.6 + 32)} = 53.35.$$

Example.—Find the volume of 15 lb. of air at 70 deg. F. under a pressure of 90 lb. per sq. in. gage.

Solution.— $PV = MRT$

$$V = \frac{15 \times 53.35 \times (460 + 70)}{144 \times (90 + 14.7)} = 28.13 \text{ cu. ft.}$$

Example.—If the air in the above problem expands to a volume of 142 cu. ft. and the final pressure is 5 lb. per sq. in. gage, what is the final temperature?

Solution.—

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

$$\frac{144 \times (90 + 14.7) \times 28.13}{(460 + 70)} = \frac{144 \times (5 + 14.7) \times 142}{T_2}$$

$$T_2 = 503.4 \text{ deg. abs.}$$

$$t_2 = 503.4 - 460 = 43.4 \text{ deg. F.}$$

14. Joule's Law: Internal Energy.—In addition to the relation between temperature, pressure, and volume for perfect gases, additional information concerning the quantity U, the internal energy, is needed. From Boyle's and Charles' laws it follows that the internal energy of a given weight of a perfect gas is independent of the volume, varying only with the temperature of the gas. In 1844, Joule, an English physicist, performed an experiment to test this deduction and found that even for a permanent gas the internal energy was essentially independent of the pressure and volume.

The apparatus used by Joule is shown in Fig. 5. The vessel A was filled with air compressed to about 300 lb. per sq. in., and the air from B was exhausted as completely as possible. The two vessels were connected by means of a pipe with a closed stop-cock C, and both vessels were immersed in a bath of water. The water in the bath was stirred and the temperature noted, then the stop-cock C was opened. After the gas had come to rest in the two vessels, the water was again stirred, and it was found that the temperature of the water in the bath had undergone no change.

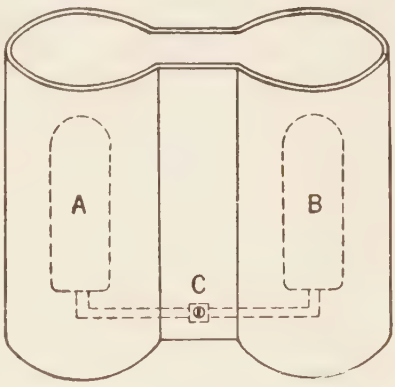


FIG. 5.—Joule's apparatus.

For an isolated gas (to which heat may not be added and on which no work can be done) it might be expected that the temperature of the gas would fall when its volume is allowed to increase, from the molecular

theory and the laws of mutual attraction. This would follow because a separation of the molecules of the gas against their mutual attraction represents an increase in the energy of the molecular system which cannot be accomplished without a decrease in the speed of the molecules. This change of speed would be indicated by a drop in the temperature. Such a transfer of energy is similar to that exhibited by a perfect pendulum in motion. Since, in the experiment described above, the water and the gas were isolated, any change in the temperature of the gas should have been indicated by a change in the temperature of the water. No such change was observed; hence it may be concluded that the mutual attraction of the molecules is so small as to be negligible, and that the total internal energy of a gas is essentially independent of the volume.

For a perfect gas exactly and a permanent gas very nearly, the *internal energy is dependent only on the temperature.* Therefore, in the energy equation $JdQ = JdU + dW$, the term dU (change of internal energy) is proportional to the change of temperature, whereas dW (external work done) depends upon the changes of pressure and volume, as explained in Art. 11.

15. Internal Energy of a Gas.—When a gas is heated through a small change of temperature dT, the heat taken in (per pound) may be expressed in the form cdT, this expression being essentially a definition of the specific heat c for the particular manner in which the heating is accomplished (see Art. 5). The symbol c_v represents the heat required to raise the temperature of one pound of the gas one degree while the volume remains constant.

When heating a given weight M of a gas at *constant volume* ($dV = 0$), the work done would be zero, and making use of the energy equation,

$$dQ = dU = Mc_v dT, \tag{17}$$

or if c_v is constant for the temperature change T_1 to T_2,

$$_1Q_2 = U_2 - U_1 = Mc_v(T_2 - T_1). \tag{18}$$

Consider next heating M pounds of a gas at *constant pressure*, from the definition of specific heat,

$$dQ = Mc_p dT, \tag{19}$$

or

$$_1Q_3 = Mc_p(T_3 - T_1) = Mc_p(T_2 - T_1), \text{ if } T_3 = T_2. \tag{20}$$

During this heating, as distinguished from the first case, some external work was done by the gas, which, according to equation 7, is

$$_1W_3 = P(V_3 - V_1). \tag{21}$$

But $PV_3 = MRT_3 = MRT_2$, if $T_3 = T_2$ and $PV_1 = MRT_1$, then equation 21 becomes

$$_1W_3 = MR(T_2 - T_1) \text{ in foot-pounds,} \tag{22}$$

or

$$_1W_3 = AMR(T_2 - T_1) \text{ in B.t.u.} \tag{23}$$

Substituting in equation 8 and transposing,

$$U_3 - U_1 = Mc_p(T_2 - T_1) - AMR(T_2 - T_1).$$

Since $T_3 = T_2$, then by Joule's law $U_3 = U_2$ and

$$U_2 - U_1 = (Mc_p - AMR)(T_2 - T_1). \tag{24}$$

Relation between c_p and c_v.—From equations 18 and 24,

$$c_v = c_p - AR, \tag{25}$$

and

$$(c_p - c_v) = AR. \tag{26}$$

For a perfect gas, both c_p and c_v would be constants and their ratio $\frac{c_p}{c_v}$ would be some constant as k, hence

$$c_p = kc_v \text{ and } c_v = \frac{c_p}{k}. \tag{27}$$

Introducing the symbol k in equation 26,

$$c_p = \frac{kAR}{(k-1)} \text{ and } c_v = \frac{AR}{(k-1)}. \tag{28}$$

Using the above relations and Joule's law the change in internal energy for *any* change of state may be expressed as follows:

$$U_2 - U_1 = Mc_v(T_2 - T_1), \tag{29}$$

or

$$U_2 - U_1 = \frac{(P_2V_2 - P_1V_1)}{(k-1)}. \tag{30}$$

The specific heats c_p and c_v of most gases have been found to increase slowly with the temperature. If the temperature range is not too large it is, however, permissible to assume constant mean values. Table I gives mean values of c_p, c_v, and k for several gases for a temperature range of 32 to 400 deg. F.

As illustrations of the results thus far obtained in the special cases when either the pressure or volume is held constant, consider the following problems.

TABLE I

MEAN SPECIFIC HEATS OF GASES

Gas	c_p B.t.u. per lb. per deg. F.	c_v B.t.u. per lb. per deg. F.	$k = \frac{c_p}{c_v}$
Air	0.242	0.173	1.40
Carbon monoxide	0.243	0.172	1.41
Carbon dioxide	0.205	0.160	1.28
Helium	1.25	0.75	1.66
Hydrogen	3.42	2.44	1.40
Nitrogen	0.247	0.176	1.40
Oxygen	0.217	0.155	1.40

Example.—Eight cubic feet of air were cooled at a constant pressure of 80 lb. per sq. in. abs. from a temperature of 180 to 100 deg. F. Find the external work done in foot-pounds, the change of internal energy in foot-pounds, and the heat abstracted in B.t.u.

Solution.— $P_1V_1 = MRT_1$,

$$80 \times 144 \times 8 = M \times 53.35 \times 640,$$

$$M = 2.7 \text{ lb. of air,}$$

From (22)

$${}_1W_2 = MR\,(T_2 - T_1),$$

$${}_1W_2 = 2.7 \times 53.35\,(560 - 640) = -11{,}520 \text{ ft-lb.}$$

NOTE.—Negative sign of W indicates compression.

From (24)

$$(U_2 - U_1) = (Mc_p - AMR)(T_2 - T_1),$$

$$(U_2 - U_1) = \left[(2.7 \times 0.242) - \frac{2.7 \times 53.35}{778}\right](560 - 640) = -37.47 \text{ B.t.u.}$$

NOTE.—Negative sign of $(U_2 - U_1)$ indicates decrease of internal energy.

From (20)

$${}_1Q_2 = Mc_p\,(T_2 - T_1),$$

$${}_1Q_2 = 2.7 \times 0.242\,(560 - 640) = -52.3 \text{ B.t.u.}$$

NOTE.—Negative sign of Q indicates removal of heat.

Example.—A tank having a volume of 25 cu. ft. contained air at a pressure of 14.5 lb. per sq. in. abs. and a temperature of 70 deg. F. How many B.t.u. were required to heat the air to a temperature of 150 deg. F.?

Solution.— $P_1V_1 = MRT_1$,

$$14.5 \times 144 \times 25 = M \times 53.35 \times 530,$$

$$M = 1.85 \text{ lb. of air.}$$

From (18)

$$_1Q_2 = Mc_v\,(T_2 - T_1),$$

$$_1Q_2 = 1.85 \times 0.173\,(610 - 530) = 25.60 \text{ B.t.u.}$$

16. Adiabatic Processes.—Another special case to which the general gas equations apply, and which is of much importance practically, is that of an expansion or compression involving no transfer of heat to or from the gas. Such a change of state is called *adiabatic* and $dQ = 0$. From this definition it follows that the work which a gas does while it is expanding adiabatically is all done at the expense of its internal energy; and the work which is spent upon a gas to compress it adiabatically goes to increase its stock of internal energy. In actual heat engines the action is never strictly adiabatic, for there are always some exchanges of heat between the working medium and the surfaces of the cylinder and piston. Rapid expansion or compression approaches being adiabatic by giving little time for heat transfer.

In the adiabatic expansion of a gas, work is done; and since no heat is taken in or given out, there must be a decrease of internal energy equivalent to the amount of work done. Taking the general energy equation, $dQ = dU + APdV$, which applies to any small change of state, $dQ = 0$ when the change is adiabatic; and hence for adiabatic expansion,

$$APdV = -\,dU, \qquad (31)$$

where $APdV$ is the work done and dU is the change of internal energy.

From equation 17, $dU = Mc_vdT$; hence from (31),

$$APdV = -\,Mc_vdT \text{ in B.t.u.}$$

But

$$P = \frac{MRT}{V} \quad \text{and} \quad \frac{AMRTdV}{V} + Mc_vdT = 0,$$

or dividing by MT,

$$\frac{ARdV}{V} + \frac{c_vdT}{T} = 0,$$

which gives by integration,

$$AR \log_e V + c_v \log_e T = \text{constant.}$$

Writing $(c_p - c_v) = AR$ from (26) and dividing by c_v, which is constant,

$$\left(\frac{c_p}{c_v} - 1\right) \log_e V + \log_e T = \text{constant}.$$

But

$$\frac{c_p}{c_v} = k,$$

thus,

$$k \log_e V - \log_e V + \log_e T = \text{constant}. \tag{32}$$

Since $\frac{PV}{T} = MR = \text{constant}$, changing to log. form,

$$\log_e P + \log_e V - \log_e T = \text{constant}. \tag{33}$$

Adding equations 32 and 33,

$$\log_e P + k \log_e V = \text{constant},$$

which gives

$$PV^k = \text{constant}, \tag{34}$$

or

$$P_1 V_1{}^k = P_2 V_2{}^k = P_3 V_3{}^k = \ldots . \tag{35}$$

Combining equations 16 and 35

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}. \tag{36}$$

In *any kind of expansion* of a gas the work done in expanding from volume 1 to volume 2 (from equation 7), is

$${}_1W_2 = \int_1^2 P dV.$$

If the nature of the expansion is such that PV^n is constant, n being any exponent, then P at *any* point is $\frac{P_1 V_1{}^n}{V^n}$, P_1 and V_1 being the pressure and volume in the initial state. Then for expansion from V_1 to V_2,

$${}_1W_2 = P_1 V_1{}^n \int_1^2 \frac{dV}{V^n},$$

which gives by integration,

$${}_1W_2 = \frac{P_1 V_1{}^n (V_2{}^{1-n} - V_1{}^{1-n})}{1 - n},$$

or

$${}_1W_2 = \frac{P_1 V_1 - P_2 V_2}{n - 1}. \tag{37}$$

Applying equation 37 to a gas expanding adiabatically, for which the exponent n assumes the value k (by equation 34),

$$ {}_1W_2 = \frac{P_1V_1 - P_2V_2}{k - 1} = \frac{MR(T_1 - T_2)}{k - 1}. \tag{38} $$

NOTE.—Equation 38 expresses also the decrease of internal energy produced by adiabatic expansion.

Example.—Three cubic feet of air enter an air compressor on the suction stroke at atmospheric pressure (14.5 lb. per sq. in. abs.) and are compressed adiabatically to a pressure of 87 lb. per sq. in. abs. (*a*) Find the work done during compression. (*b*) What was the final temperature, if the initial temperature was 68 deg. F.?

Solution.—(*a*) From (35),

$$ V_2 = 3\left(\frac{14.5}{87}\right)^{\frac{1}{1.4}} = 0.834 \text{ cu. ft.} $$

From (38),

$$ {}_1W_2 = \frac{144\,[(14.5 \times 3) - (87 \times 0.834)]}{0.40} = -\,10{,}461 \text{ ft-lb.} $$

(*b*) From (36),

$$ T_2 = 528\left(\frac{87}{14.5}\right)^{\frac{0.4}{1.4}} = 880.9 \text{ deg. abs.} $$

$$ t_2 = 420.9 \text{ deg. F.} $$

17. Isothermal Processes.—Still another special case, to which the general gas equations apply and which is of practical importance, is that of an expansion or compression during which the temperature of the gas remains constant. Such a change of state is called *isothermal.* When a gas is expanding isothermally, heat is added to maintain its temperature constant; the gas, therefore, must be in contact with an external source of heat. When it is being compressed isothermally the gas gives out heat, and must be in contact with a receiver which can take heat from the gas; the water jacket of an air compressor serves as a partial receiver of heat. If T is made constant in the equation $PV = MRT$,

$$ PV = \text{constant}, \tag{39} $$

or

$$ P_1V_1 = P_2V_2 = P_3V_3 = \ldots . \tag{40} $$

This is the equation of the isothermal curve in PV coordinates, and the curve is an equilateral hyperbola. Figure 6 shows the isothermal (ABC) and adiabatic ($A'BC'$) curves drawn on the PV plane.

To find the external work done in the isothermal expansion of a gas from V_1 to V_2,

$$ {}_1W_2 = \int_1^2 P dV, $$

and from (39)

$$ P = \frac{P_1 V_1}{V}, $$

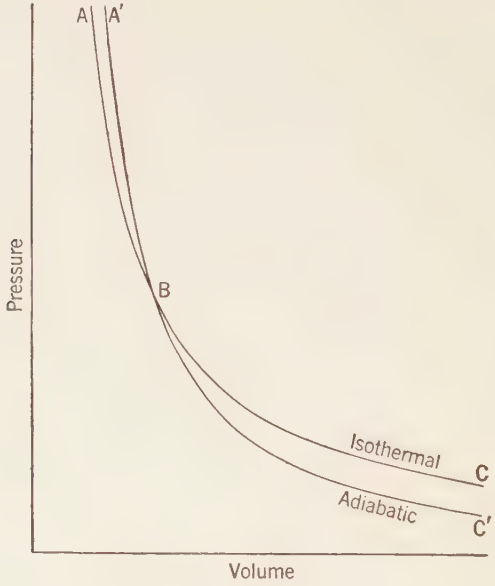


FIG. 6.—Pressure-volume curves.

from which

$$ {}_1W_2 = P_1 V_1 \int_1^2 \frac{dV}{V}. $$

Integrating

$$ {}_1W_2 = P_1 V_1 (\log_e V_2 - \log_e V_1), $$

or

$$ {}_1W_2 = P_1 V_1 \log_e \frac{V_2}{V_1}. \tag{41} $$

But since

$$ PV = P_1 V_1 \quad \text{and} \quad PV = MRT; $$

then

$$ {}_1W_2 = MRT \log_e \frac{V_2}{V_1}, \tag{42} $$

where T is the absolute temperature at which the process takes place. This expression gives either the work that is done by a gas in isothermal

expansion, or the work that is spent on the gas in isothermal compression.

During the isothermal expansion or compression of a perfect gas there is no change of internal energy, since there is no change of temperature (Art. 14). Hence, during isothermal expansion a gas must take in an amount of heat equivalent to the work done, from equation 9, $dQ = APdV$, and from equation 42, ${}_1Q_2 = AMRT \log_e \frac{V_2}{V_1}$; and during isothermal compression from V_2 to V_1 it must reject that amount of heat.

Example.—Three cubic feet of air under a pressure of 80 lb. per sq. in. abs. were expanded isothermally to a pressure of 20 lb. per sq. in. abs. (*a*) Find the work done during this expansion. (*b*) Find the heat added in B.t.u.

Solution.—(*a*) From (40),

$$V_2 = \frac{3 \times 80}{20} = 12 \text{ cu. ft.}$$

From (41),

$${}_1W_2 = 144 \times 80 \times 3 \log_e \tfrac{1}{3} = 47{,}910 \text{ ft-lb.}$$

(*b*) Heat added $= \frac{47{,}910}{778} = 61.6$ B.t.u.

18. Reversible and Irreversible Processes.—When considering thermodynamic processes in which a working medium may be expanded or compressed, and may take in or give out heat, the conception of reversibility plays an important part. There are no reversible processes in nature; but the limiting cases, when resistance either to motion or the flow of heat is considered reduced indefinitely to a minimum, may be analyzed as if such no-resistance cases actually existed. An expansion or compression is *reversible* if the entire process and order of events can be carried out in the reverse order, and the medium returned to its original state without any change in the medium or in any other of the connected systems. An *irreversible* process is one which cannot be reversed without leaving a change in some connected system when the medium is brought to its original state.

A reversible mechanical process is frictionless; the expansion of a gas, to be reversible, must take place with no loss of mechanical effect because of internal molecular friction. A reversible thermal process is one in which heat will flow from one substance to another without a difference in temperature existing between the two substances, a condition approached as the difference in temperature is made smaller and smaller by allowing an infinitely long time for the flow of heat.

Friction makes any mechanical process irreversible; friction of the flow of a gas from a region of high pressure to a lower one, as introduced in Joule's experiment, is an example of irreversible flow. Another irreversible process is exemplified when two substances of different temperatures are placed in contact with each other and heat flows from one to the other.

Reversible processes are used in the study of an ideal engine to derive an expression for its maximum efficiency, and further to determine the factors, exclusive of friction, which influence the magnitude of the maximum efficiency. Experiment and experience alone can be used to evaluate the effect of friction on the actual efficiency of the real engine.

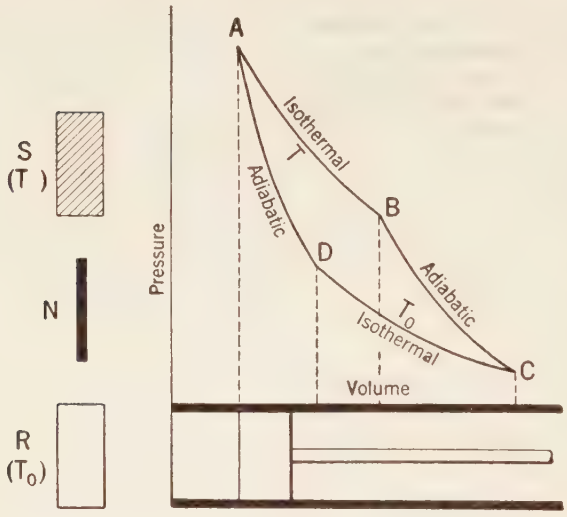


FIG. 7.—The Carnot cycle, pressure-volume plane.

19. The Carnot Cycle.—Carnot, a French engineer, pointed out (in 1824) that the greatest possible amount of work may be obtained by letting the heat pass from the source to a receiver, through an engine working in a reversible manner—reversible not only as regards its own internal actions but also as regards the transfer of heat to the engine from the source, and from the engine to the receiver. The engine described by Carnot is an engine each of whose operations is reversible as explained in Art. 18.

The Carnot cycle for a perfect gas, Fig. 7, consists of four successive operations: (1) isothermal expansion, heat taken in at the temperature of the hot source T; (2) adiabatic expansion, during which the temperature of the working medium falls from T to T_0; (3) isothermal compression, heat rejected at the temperature of the receiver T_0; (4) adiabatic compression, during which the temperature of the working medium is raised from T_0 to T. This completes the cycle by bringing the medium back to the condition in which it was assumed to be at the beginning of the first operation.

Consider an ideal engine in which a working medium may go through the operations of Carnot's cycle, Fig. 7. Imagine a cylinder and

piston composed of non-conducting material, except the end of the cylinder which is a conductor. A hot body S serves as an infinite source of heat, kept at temperature T; N is a non-conducting cover; and R is a cold body of infinite capacity for receiving heat, kept at temperature T_0 which is lower than T. Suppose that S, N, or R can be applied to the end of the cylinder, as desired.

Throughout this chapter the properties of perfect gases have been used to illustrate some of the general thermodynamic principles, because of the ease with which these properties may be described analytically. Suppose, therefore, that the working medium used in the ideal Carnot engine is a perfect gas. Let the cylinder contain M pounds of a perfect gas at temperature T, volume V_a, and pressure P_a. Apply the hot source S to the conductor end of the cylinder and allow the frictionless piston to advance slowly to V_b and P_b. The gas expands isothermally at T, taking in heat and doing work. The non-conducting cover N is next placed against the cylinder end, and the gas is expanded adiabatically to V_c and P_c; the piston advances and does work at the expense of the internal energy of the gas, while the temperature falls to T_0. Then the cold body of receiver R is applied to the cylinder end, the piston is forced back slowly and the gas is compressed isothermally at T_0, since the smallest increase of temperature above T_0 causes heat to pass into R. Work is spent upon the gas and heat is rejected to the receiver R. Let this rejection of heat be continued until point D is reached, such that the next operation will complete the cycle. Remove R and apply N, continue the compression which is now adiabatic. The pressure and temperature rise, and, if point D has been properly chosen, when the pressure is restored to its original value P_a, the temperature will also have risen to its original value T. This completes the cycle.

20. Efficiency of the Carnot Cycle.—During the adiabatic expansion from B to C, Fig. 7, the relation between volume and temperature, from equation 36 is

$$\left(\frac{V_c}{V_b}\right)^{k-1} = \frac{T_b}{T_c} = \frac{T}{T_0};$$

and during the adiabatic compression from D to A, the relation is

$$\left(\frac{V_d}{V_a}\right)^{k-1} = \frac{T_a}{T_d} = \frac{T}{T_0},$$

hence

$$\frac{V_c}{V_b} = \frac{V_d}{V_a} \quad \text{or} \quad \frac{V_b}{V_a} = \frac{V_c}{V_d}. \tag{43}$$

The transfers of heat in the four successive stages of the Carnot cycle are:

(1) Heat taken in from S, $_aQ_b = MRT \log_e \frac{V_b}{V_a}$. (See Art. 17.)

(2) No heat taken in or rejected.

(3) Heat rejected to R, $_cQ_d = MRT_0 \log_e \frac{V_c}{V_d}$. (See Art. 17.)

(4) No heat taken in or rejected.

The area $ABCD$, Fig. 7, enclosed by the four curves of the cycle, represents the mechanical work done. The net amount of external work done by the gas is the difference between the heat taken in and the heat rejected in a complete cycle. Then from the first law of thermodynamics, $(Q) = {}_aQ_b - {}_cQ_d = (W)$. The *efficiency of the cycle* is the ratio of the work gained to the heat supplied from the source. Denoting the Carnot cycle efficiency by e_c,

$$e_c = \frac{(Q)}{{}_aQ_b} = \frac{(W)}{{}_aQ_b} = \frac{MRT \log_e \frac{V_b}{V_a} - MRT_0 \log_e \frac{V_c}{V_d}}{MRT \log_e \frac{V_b}{V_a}}.$$

But from (43),

$$\frac{V_b}{V_a} = \frac{V_c}{V_d};$$

then

$$e_c = \frac{T - T_0}{T}. \tag{44}$$

Suppose that there are two heat engines A and B, Fig. 8, one of which (A) is reversible, working between the same hot source of heat and cold receiver, and each producing the same amount of mechanical work. Let Q be the quantity of heat which A takes in from source S. Now if A is reversed, this engine will, by the expenditure on it of the same amount of work, give to the hot source the amount of heat formerly taken from S, namely Q. For this purpose set the engine B to drive A reversed; the work which B produces is just sufficient to drive A, and the two machines (B driving A) form together a self-acting machine unaided by any external agency. Engine B takes heat from the hot body, and A, which is reversible, gives back to S the amount of heat Q. Now if B could do its work by taking less heat than Q from S, the hot body would gain heat. But no work is being done on the system from the outside, nor is any heat supplied from other sources, so whatever heat the hot body gains must come from the cold body. Therefore, if engine B could

do as much work as the reversible engine A, with a smaller supply of heat, it would be possible to arrange a self-acting machine through which heat would pass from a cold body to a hot body, thus violating the second law of thermodynamics.

Thus engine B cannot do the same amount of work with a smaller supply of heat than a reversible engine; or, *no other engine can be more efficient than a reversible engine, when they both work between the same two temperatures of source and receiver.* If both engines A and B were considered reversible, the same argument shows that they are equally efficient. Hence, all reversible engines working between the same temperature limits are equally efficient, which necessarily implies that *the efficiency is independent of the working medium.*

Equation 44, $e_c = \dfrac{T - T_0}{T}$, measures the efficiency of *any* reversible heat engine; and therefore also expresses the largest fraction of the heat supplied that can possibly be converted into work by any engine when operating between temperature limits, T and T_0. If there is a supply of heat at a temperature T and a means of disposing of heat at a minimum temperature T_0, then there is no possibility of converting more than a fraction of the heat into work. This is the measure of the *best efficiency;* it is the theoretical limit beyond which the efficiency of a heat engine cannot go. No engine can surpass this standard; and the efficiency of a real engine is always less, as no real engine is strictly reversible.

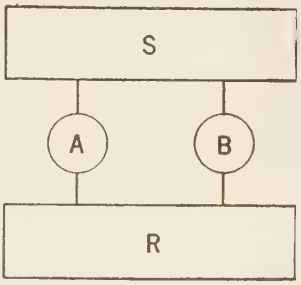


FIG. 8.— Two heat engines with source and receiver.

21. Entropy.—The *availability* of a given quantity of heat energy for transformation into work is given by the efficiency of the ideal Carnot engine. Thus, if T is the temperature of the source of heat and T_0 the temperature of the coldest receiver available, then the efficiency is, $e_c = \dfrac{T - T_0}{T} = 1 - \dfrac{T_0}{T}$. If the heat dQ has been removed from the source, then $\left[1 - \left(\dfrac{T_0}{T}\right)\right] dQ$ has been transformed into work under ideal conditions, but the part $\left(\dfrac{T_0}{T}\right) dQ$ would be rejected. Consider any thermodynamic system undergoing any desired change of state; divide its internal energy at any stage of the process into two parts which may be called the available part and the unavailable part, respectively. If, during the process considered, an amount of heat dQ

is transferred to the system, the part $T_o\left(\frac{dQ}{T}\right)$ of this heat is unavailable and may be supposed to increase the unavailable energy of the system by this same amount. In general, unless the volume of the system is held fixed, mechanical work is done at the same time that heat is being added. But mechanical work has been used as available energy in all the considerations of the Carnot cycle and hence will not affect the *unavailable* part of the internal energy.

Dividing the increase of unavailable energy by T_0, a definition of what is called the entropy of the system is obtained. Thus,

$$dS = \frac{dQ}{T}. \tag{45}$$

This equation is applicable for reversible processes only, as every process in the Carnot cycle is reversible; and the efficiency of the ideal Carnot engine was used as the underlying principle in the preceding paragraphs. In a frictionless adiabatic process no heat is added or rejected and $\frac{dQ}{T}$ becomes zero from equation 45, hence the entropy does not change. The direction of a process, physical or chemical, that occurs of itself is such as will bring about an increase of unavailable energy of the system. This principle is the foundation of the application of thermodynamics to chemistry. Electrical energy is rendered unavailable when transformed into heat in the transmission system. In the transformation of work into electrical energy or of electrical energy into mechanical work, some of the high-grade energy is transformed into heat and is rendered unavailable.

Experience shows that every actual physical process is irreversible and accompanied by losses. As a result the actual or irreversible process is accompanied by a decrease of the quantity of energy available for transformation into work, or an increase in unavailable energy. For an irreversible process the change of entropy is

$$dS = \frac{dQ}{T} + \frac{dQ'}{T}, \tag{46}$$

in which dQ denotes the heat added by the medium from outside and dQ' the heat generated within the medium because of friction.

When a thermodynamic system undergoes any change, the sum of the entropy changes of the bodies which take part is the change in entropy of the system. The entropy of a system is the same whenever the system is in the same physical state. When a medium has passed through any complete cycle of operations its entropy at the end is

identical with the entropy at the beginning, for the original state has been restored in all respects. This is true of an irreversible cycle as well as for a reversible cycle. Entropy, like energy, is relative, and any expression for the entropy of a system contains an unknown constant of integration; but since only differences of entropy are considered, the constant of integration disappears by subtraction. The entropy changes of a medium are usually calculated per unit of weight, and numerical values of it are given in tables of the properties of steam and of other mediums used in heat engineering. In assigning these numerical values it is customary to assume the entropy as zero at some standard state. Thus, in the tables of properties of steam, the entropy of the liquid at 32 deg. F. is taken as zero.

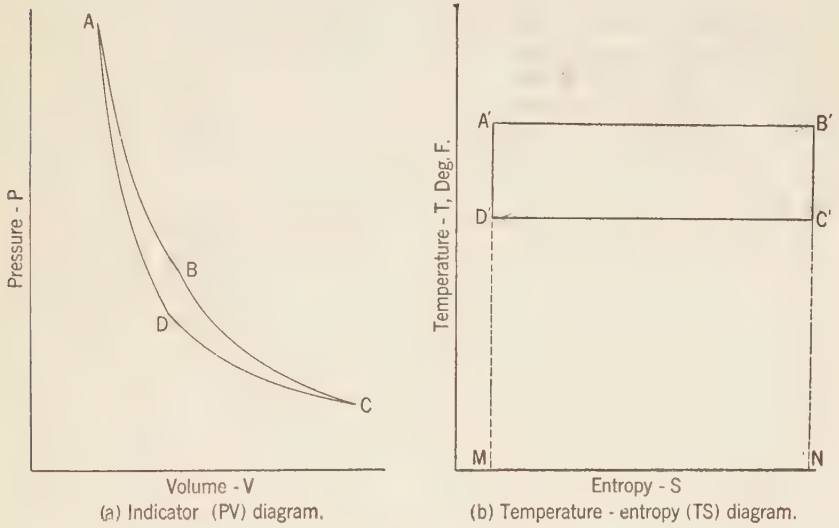


FIG. 9.—Carnot cycle for air.

22. Temperature-Entropy Diagrams.—In making use of entropy, it may be considered for reversible changes as a quantity having the property that the heat absorbed by a thermodynamic system is given by the expression

$$dQ = TdS,$$

just as the mechanical work performed is

$$dW = PdV.$$

It was shown in Art. 12 that the area under a curve on the PV plane represented the work done for a reversible change of pressure and volume. Of equal interest is the graphical representation of the changes

of entropy on a plane having the absolute temperature as the ordinate and the entropy changes as the abscissa; for a reversible process, the area under a curve on the TS plane represents the heat added or rejected. If a complete cycle were drawn on the PV plane the same cycle may be described on the TS plane, and if all the changes were reversible the enclosed areas would represent the work done around the cycle. Since values of entropy are expressed usually in B.t.u. per degree per pound the area on the TS plane will represent the work in B.t.u., whereas the area on the PV plane is generally foot-pounds. Figures 9*a* and 9*b* show a Carnot cycle for air. In each diagram the work done is drawn in foot-pounds except that the area in Fig. 9*b* has been made $2\frac{1}{2}$ times the area in Fig. 9*a* for convenience of illustration. If in Fig. 9*a* one unit of area represents one foot-pound and in Fig. 9*b* one unit of area represents one B.t.u. the enclosed area on Fig. 9*b* should be $\frac{1}{778}$ of the enclosed area on Fig. 9*a*.

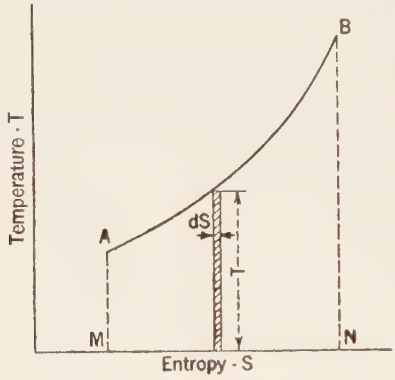


FIG. 10.—Temperature-entropy curve.

If curve AB in Fig. 10 represents a reversible change of state, it can be shown that the area under this curve represents the heat absorbed by the medium if the change takes place from A to B or the heat rejected if the change occurs from B to A. The area under the curve is the sum of all the elementary areas TdS taken from A to B, which may be written $\int_A^B TdS$. But $dQ = TdS$ for a reversible change of state, hence the area $MABN$ under the curve represents the heat absorbed.

PROBLEMS

1. An air receiver had a volume of 125 cu. ft. Find the weight of air it contained if the temperature was 70 deg. F. at a pressure of 110 lb. per sq. in. abs.

2. (*a*) What volume would 6 lb. of hydrogen occupy at standard atmospheric pressure and 100 deg. F.? (*b*) Find the temperature of 28 lb. of oxygen for a volume of 5 cu. ft. at a pressure of 1000 lb. per sq. in. abs.

3. The gas constant R for methane is 96.34 and the value of k is 1.31. Calculate the specific heats c_p and c_v for this gas.

4. Fifty cubic feet of air at a pressure of 120 lb. per sq. in. gage and at a temperature of 250 deg. C. were expanded to a condition of 89 cu. ft. and 285 deg. F. Find the weight of the air and resulting pressure after expansion. Barometer was 29.5 in. of mercury.

5. An air hammer required 0.85 cu. ft. of air at 50 lb. per sq. in. gage pressure for each operation. The air was supplied from a storage tank containing 100 cu. ft. of air at 250 lb. per sq. in. gage pressure. How many times did the hammer operate? Barometer, 29.18 in. of mercury. Assume constant air temperature.

6. A fan delivered 20,300 cu. ft. of air at a temperature of 90 deg. F. and an absolute pressure of 29.3 in. of mercury. What was the volume at standard conditions of 60 deg. F. and 29.92 in. of mercury?

7. A manufacturer contracted to buy natural gas at 20 cents per thousand cubic feet when the gas was measured at 14.7 lb. per sq. in. abs. and 60 deg. F. How much did he pay for 56,420 cu. ft. if it was measured at 8 in. of water gage pressure and 90 deg. F.? Barometer, 29.42 in. of mercury.

8. The volume of a tank was 30 cu. ft.; it contained air under a pressure of 100 lb. per sq. in. abs. at a temperature of 75 deg. F. Some of the air escaped until the pressure of the air in the tank was reduced to 60 lb. per sq. in. abs. and the temperature was 65 deg. F. Find the volume occupied by the air removed from the tank at standard atmospheric pressure (29.92 in. mercury) and 60 deg. F.

9. Ten cubic feet of air at a temperature of 60 deg. F. were entrapped below a piston in a vertical cylinder. The piston was 10 in. in diameter and weighed 500 lb. If 400 lb. were added to the weight of the piston, calculate the distance in inches the piston moved if there was no friction and the temperature of the air remained constant.

10. A closed tank contained 100 cu. ft. of air at a temperature of 75 deg. F. and a pressure of 95 lb. per sq. in. abs. (*a*) Find the weight of air. (*b*) How much heat was required to raise the temperature of the air to 110 deg. F. and what was the change in internal energy? (*c*) Find the pressure in the tank after heating. (*d*) How much external work was done in part *b*?

11. The combustion of one pound of oil added 18,000 B.t.u. to a quantity of air at constant pressure. If the temperature of the air was raised from 70 to 2600 deg. F.: (*a*) Find the weight of the air. (*b*) What was the external work done in ft-lb.? (*c*) Find the change in internal energy in B.t.u.

12. How many B.t.u. were absorbed by 8 lb. of air when expanded to double its initial volume at a constant temperature of 80 deg. F.?

13. Five cubic feet of air were expanded from 200 to 50 lb. per sq. in. abs. in a cylinder where the temperature was kept constant by a water jacket. (*a*) Find the external work done in foot-pounds. (*b*) Compute the heat added during expansion. (*c*) What was the change in internal energy?

14. An air compressor was arranged to compress 200 cu. ft. of air from 14.5 to 120 lb. per sq. in. abs. at constant temperature. (*a*) Compute the external work performed. (*b*) Find the heat removed from the air during compression.

15. During the compression of air, 20 B.t.u. were removed by the cooling water in the jacket. If at the same time 95,400 ft-lb. of external work were done upon the air, find the change in internal energy.

16. Forty B.t.u. were added to 10 lb. of air. The initial pressure of the air was 60 lb. per sq. in. abs. at a temperature of 70 deg. F. (*a*) Find the resulting temperature, if the heat was added at constant volume. (*b*) What would have been the resulting temperature and the external work done, if the heat had been added at constant pressure?

17. The piston displacement volume of an air compressor was 1.85 cu. ft. It took in air at 15 lb. per sq. in. abs. and 70 deg. F., and compressed it isothermally to 115 lb. per sq. in. abs. (*a*) Find the weight of air in the cylinder at the beginning of the compression stroke. (*b*) What was the volume of the air at the end of compression? (*c*) Find the work of compression in foot-pounds. (*d*) How much heat

was rejected by the air during compression? (*e*) What was the change in internal energy in B.t.u.?

18. Using the same data as in problem 17, assume the compression to have been adiabatic. (*a*) Find the volume of the compressed air. (*b*) What would be the temperature of the air at the end of compression? (*c*) Find the work of compression in foot-pounds. (*d*) How much heat would be rejected by the air during compression? (*e*) What would be the change in internal energy in B.t.u.?

19. With the same data as given in problem 18, assume the exponent for the compression curve to have been 1.28. (*a*), (*b*), (*c*), (*d*), and (*e*) as in problem 18.

20. Two cubic feet of air at 60 deg. F. and an initial pressure of 14.5 lb. per sq. in. abs. were compressed in a cylinder to 87 lb. per sq. in. abs. If no transfer of heat occurred, find the final temperature and volume.

21. If 1.5 cu. ft. of air at 200 lb. per sq. in. abs. were expanded to a pressure of 25 lb. per sq. in abs., calculate the final volume and the external work done: (*a*) if the expansion was isothermal, and (*b*) if the expansion was adiabatic.

22. An air compressor delivered 20 cu. ft. of air per minute at a pressure of 120 lb. per sq. in abs. The initial volume was 100 cu. ft. at a pressure of 15 lb. per sq. in. abs. (*a*) Find the value of the exponent n. (*b*) What change in temperature accompanied this compression? (*c*) Find the work of compression. (*d*) How much heat was added or rejected per minute?

23. An ideal engine operating on the Carnot cycle contained 2 lb. of air as the working medium. The pressure at the beginning of isothermal expansion was 120 lb. per sq. in. abs. and the volume was 4 cu. ft. The exhaust temperature was 40 deg. F. Assume that the heat added during the cycle was 10 B.t.u. (*a*) Find the efficiency of the cycle. (*b*) What was the external work of the cycle in foot-pounds?

24. A heat engine was operated between the temperature limits of 700 and 110 deg. F. The engine was supplied with 16,350 B.t.u. per brake horsepower per hour. (*a*) What was the actual thermal efficiency? (*b*) Find the Carnot or maximum theoretical efficiency.

25. Assume a temperature range of 200 deg. F. for a Carnot cycle during which 50 B.t.u. were added to 3 lb. of air at a temperature of 500 deg. F. (*a*) What was the efficiency of the cycle? (*b*) How much heat was rejected during the cycle? (*c*) Find the external work done in foot-pounds. (*d*) Determine the change in internal energy along one adiabatic curve.

26. In a Carnot cycle with air as the working medium, refer to Fig. 7, the pressure p_a was 135 lb. per sq. in. abs. at 360 deg. F., and the pressures p_b and p_c were 75 and 15 lb. per sq. in. abs., respectively. The volume V_a was one cubic foot. (*a*) Find the temperatures t_b, t_c, and t_d. (*b*) Make a sketch of the cycle and identify the changes with proper values of p, V, and T. (*c*) Compute the external work done of the cycle in foot-pounds. (*d*) What was the change in internal energy? (*e*) Determine the efficiency of the cycle.

27. The volume of a quantity of air was 10 cu. ft. at 60 deg. F. and 20 lb. per sq. in. abs. pressure; heat was added at constant volume until the pressure was 150 lb. per sq. in. abs. It was then expanded according to $PV^n = C$ to a pressure of 20 lb. per sq. in. abs. and a volume of 75 cu. ft. The air was finally compressed at constant pressure to original conditions. Find: (*a*) The value of n. (*b*) The temperature at the end of each step. (*c*) The heat added. (*d*) The heat rejected. (*e*) The external work done. (*f*) The efficiency of the cycle.

28. Air was used as the working medium for a closed cycle in which p_1 was 15 lb. per sq. in. abs. and V_1 was one cubic foot. The gas was compressed isothermally to a pressure of 75 lb. per sq. in. abs., the volume being decreased to 0.20 cu. ft.

Heat was added at constant volume until the pressure increased to 225 lb. per sq. in. abs. The air was then expanded adiabatically to a pressure of 15 lb. per sq. in. abs., after which heat was rejected at constant pressure until the volume was reduced to one cubic foot. Find: (*a*) The heat added in B.t.u. (*b*) The heat rejected in B.t.u. (*c*) The external work done per cycle. (*d*) The efficiency of the cycle.

29. A gas engine used an air and gas mixture, for which $c_p = 0.25$ and $c_v = 0.19$. One-fourth of a cubic foot was taken into the cylinder at a temperature of 100 deg. F. and a pressure of 14 lb. per sq. in. abs. This gas mixture was (1) compressed adiabatically to a temperature of 450 deg. F., (2) heated at constant volume until the temperature rose to 1500 deg. F., (3) expanded adiabatically to a volume of 0.25 cu. ft., and (4) cooled at constant volume until the original conditions of temperature and pressure were attained. (*a*) Find the weight of air and gas mixture used per cycle. (*b*) Draw a PV diagram. (*c*) What was the heat added in B.t.u.? (*d*) Determine the efficiency of the cycle. (*e*) What horsepower was being developed by the engine if there were 800 cycles per minute?

30. A quantity of air whose volume was 3.02 cu. ft. at a pressure of 14.7 lb. per sq. in. abs. at a temperature of 60 deg. F. was (1) compressed adiabatically to a pressure of 500 lb. per sq. in. abs. and a temperature of 980 deg. F., (2) expanded along a constant pressure line to a volume of 0.753 cu. ft., (3) expanded adiabatically to a pressure of 71.6 lb. per sq. in. abs. and a volume of 3.02 cu. ft., and (4) returned to the original condition along a constant volume path. (*a*) What was the volume after being compressed to 500 lb. per sq. in. abs. and 980 deg. F.? (*b*) Draw a PV diagram. (*c*) Determine the heat added. (*d*) What was the external work done in foot-pounds? (*e*) Calculate the efficiency of the cycle.

CHAPTER III

HEAT POWER PLANTS

23. Power Production.—Mechanical power, as commercially produced, is derived from heat energy (liberated by burning some fuel) or by the action of flowing water. As only consideration of heat power plants falls within the scope of this book, hydraulic or water power plants will not be discussed. In the normal production of power, heat energy contained in the fuel is constantly supplied at one end of the line of plant apparatus. Mechanical or electrical power is delivered at the other end of the system. The transformation of energy is accomplished in either *steam plants* or *internal-combustion engine plants*. Both types of plants require a complete assembly of equipment to produce the desired change.

24. Prime Movers.—The principal machine of a power plant which converts heat energy into mechanical energy is known as the *prime mover*. Steam plants employ as prime movers either turbines or reciprocating engines or both. The steam used is generated in a boiler or boilers by burning the fuel in a furnace or furnaces which are an integral part of the boiler setting. The steam formed by the vaporization of water in the boiler serves as a heat carrier, which is readily conveyed through pipe lines to the prime movers. Other apparatus such as pumps, heaters, condensers, stokers, fans, etc., may be required for the proper operation of the plant. The prime movers of steam plants may be operated either *non-condensing* or *condensing*.

In non-condensing operation the steam exhausted from the prime movers is discharged at atmospheric pressure or at pressures greater than atmospheric. Condensing-plant prime movers exhaust their discharge steam into a condenser in which the pressure is less than atmospheric, and in which the steam is reduced to water. In neither non-condensing nor condensing operation is a prime mover (using steam) able to abstract enough of the heat energy carried by the steam to reduce it completely to water. The principal advantages of condensing operation are the increased amount of energy abstracted per pound of steam, and the greater amount of power that may be developed by a given size of engine or turbine.

Internal-combustion engine plants operate with the combustion of the fuel taking place within the prime mover. The fuels commonly

used by internal-combustion engines are: natural gas, producer gas, blast-furnace gas, coke-oven gas, kerosene, gasoline, crude petroleum or its distillates. These plants usually show a better utilization of the heat energy of the fuel, as a number of thermal losses are avoided. When a selection of internal-combustion engines as the prime movers of a plant is considered, attention must be given to the following items: initial cost, maintenance charges, depreciation, and the limited amount of power that may be developed on a given floor space.

Dependent upon the fuel used, internal-combustion engines require a certain amount of auxiliary apparatus for their proper operation. The auxiliary apparatus to be included in the plant may be: fuel pumps, fuel heaters, storage tanks, gas producers, gas coolers, gas scrubbers, tar extractors, and gas holders.

25. Cycle of Operation of a Non-condensing Plant.—A sectional elevation of a small non-condensing steam power plant is illustrated by Fig. 11. Fuel is introduced into the boiler furnace by means of an underfeed stoker. The necessary air for combustion flows from the stoker windboxes up through tuyères into and through the fuel bed and produces burning of the fuel. The hot gaseous products of combustion rise from the furnace and move through the various passes of the boiler setting into the breeching, at the rear, through which they are carried to the stack or chimney. In this manner the hot gases are brought into contact with a considerable amount of boiler heating surface. A part of the heat liberated by the combustion of the fuel is transferred through the heating surface to heat and evaporate water supplied to the boiler. Not all the heat content of the fuel is utilized in the boiler, as there are certain inevitable losses. The losses include: heat carried away by the flue gases in the stack, unburned volatile constituents, radiation, and unburned fuel in the ash and refuse.

The steam produced in the boiler flows from the boiler through piping and a steam separator to the engine. The function of the separator is to remove the greater portion of any water particles that may be carried by the steam. The separated water may be returned by a trap to the boiler feed supply in the feedwater heater. The dried steam is expanded in the two cylinders of the engine, where heat energy of the steam is converted into mechanical energy, as the steam acts against the engine pistons. The power developed by the engine is transmitted through the engine crankshaft to an electric generator, where the conversion into electrical energy is made. The major portion of the electrical energy of the generator is delivered, as the plant output, at the switchboard.

The expanded steam is exhausted from the engine at a reduced pressure and temperature. The exhaust steam is carried to a feedwater

heater, where a portion of it is utilized in heating the water supplied to the boiler. A feed pump is used to force the heated water into the

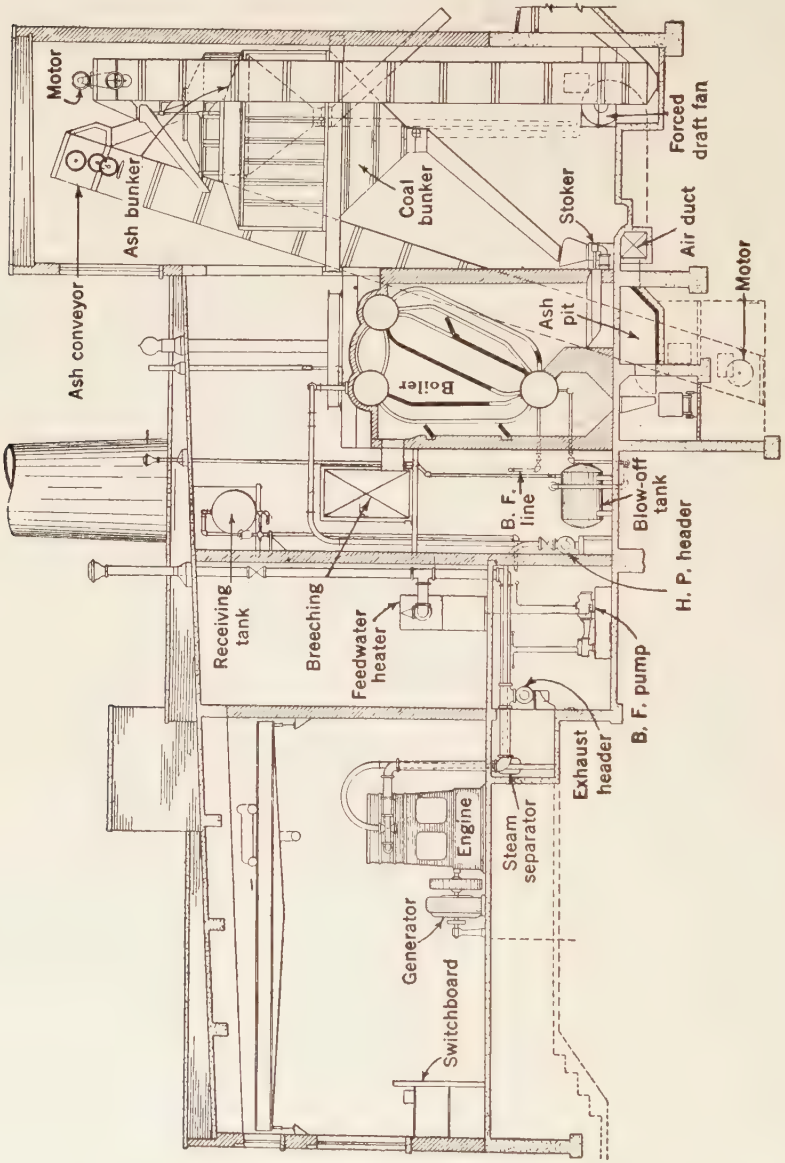


FIG. 11.—Sectional elevation of a non-condensing steam power plant.

boiler. Exhaust steam not used by the water heater may be utilized in the radiators of a heating system or in manufacturing processes, or it may be wasted to the atmosphere.

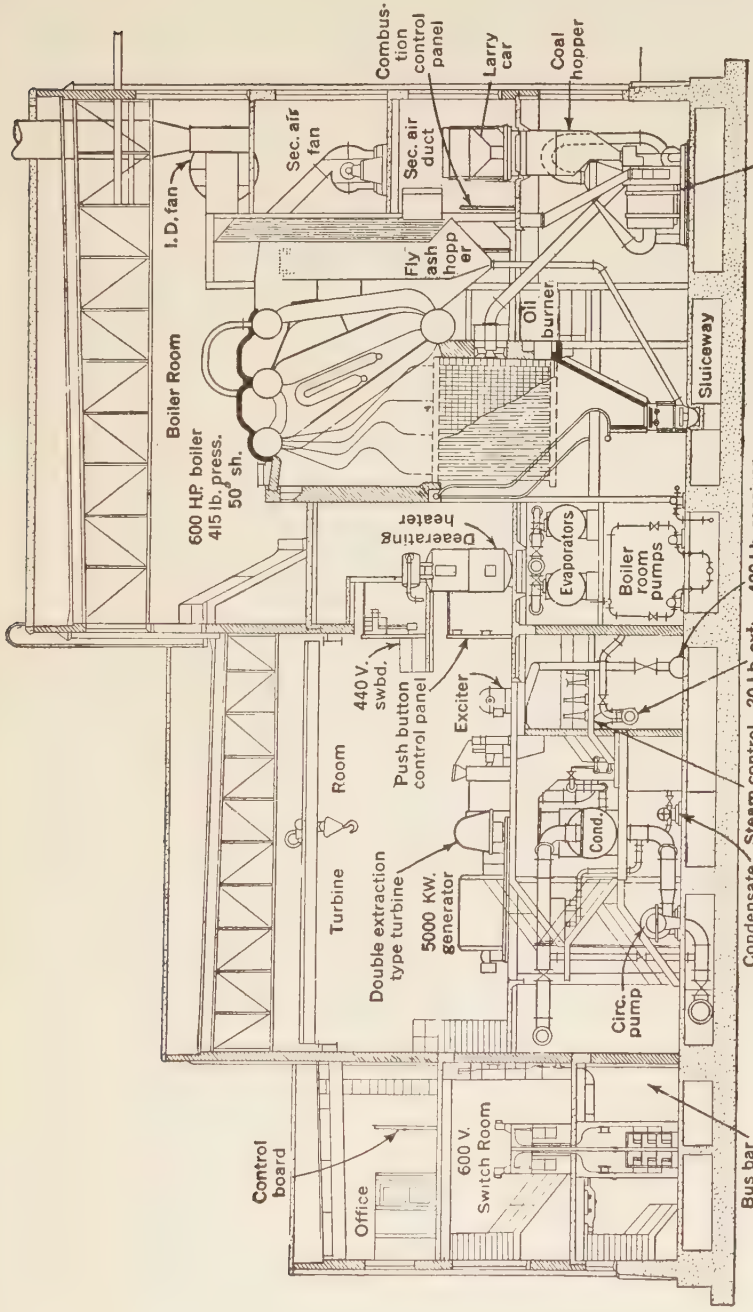


FIG. 12.—Sectional elevation of a condensing steam power plant.

From the coal pile to the switchboard, in each step of the process and in each machine, there are thermal and mechanical losses which reduce the amount of the available energy that is delivered as useful output. The useful output delivered by non-condensing plants may be 3 to 10 per cent of the energy contained in the fuel when the exhaust steam is discharged to the air. If the excess exhaust steam is used for heating, the percentage of the total heat of the fuel utilized in power production and heating may be increased.

26. The Condensing Plant Cycle.—Figure 12 illustrates a modern steam turbine plant in which considerable auxiliary apparatus is installed to conserve and utilize the heat energy of the fuel. The fuel used may be either oil or powdered coal. The plant is equipped with a condenser, and differs from the non-condensing plant in the matter of exhaust steam disposal. In condensing operation, the steam discharged from the prime mover is reduced to water in the condenser. This process produces a considerable reduction of the back pressure against which the prime mover works and promotes a better economy in the use of steam and fuel. All, a part, or none of the condensed steam may be returned to the boilers, depending upon the type of condensers used.

The overall efficiency, or ratio of the plant heat output as work to the heat input, ranges from 7 to 28 per cent for condensing plants. The latter figure is obtainable only in a very few large modern stations, in which every effort is made to secure exceedingly efficient operation.

27. Internal-combustion Engine Plant.—Plants using internal-combustion engines may burn any of the fuels listed in Art. 24, provided that the engine is properly designed to use the fuel. The plant shown in plan and elevation by Fig. 13 employs a Diesel internal-combustion engine to drive an electric generator. The principal parts of the plant are: starting device A; bottles for storing compressed air used in engine starting B; barring gear D; lubricating oil pumps and filter tanks OF; fuel filters for both heavy and light oils HF and LF; overhead lubricating oil tanks and coolers S; fuel storage tank T; exhaust piping P; and exhaust muffler M.

The working medium of any internal-combustion engine is the products of combustion formed by burning a fuel within one or both ends of its one or more cylinders. The fuels used may be in a gaseous state originally; they may be light liquid fuels gasified by evaporation; or they may be heavy fuel oils injected into the cylinders of Diesel engines by means of a fuel pump or an air-jet. Units to use any of the foregoing fuels may operate by requiring either two strokes (two-cycle)

or four strokes (four-cycle) of the engine piston to complete a cycle within a cylinder end, and are suitable for use as prime movers. An

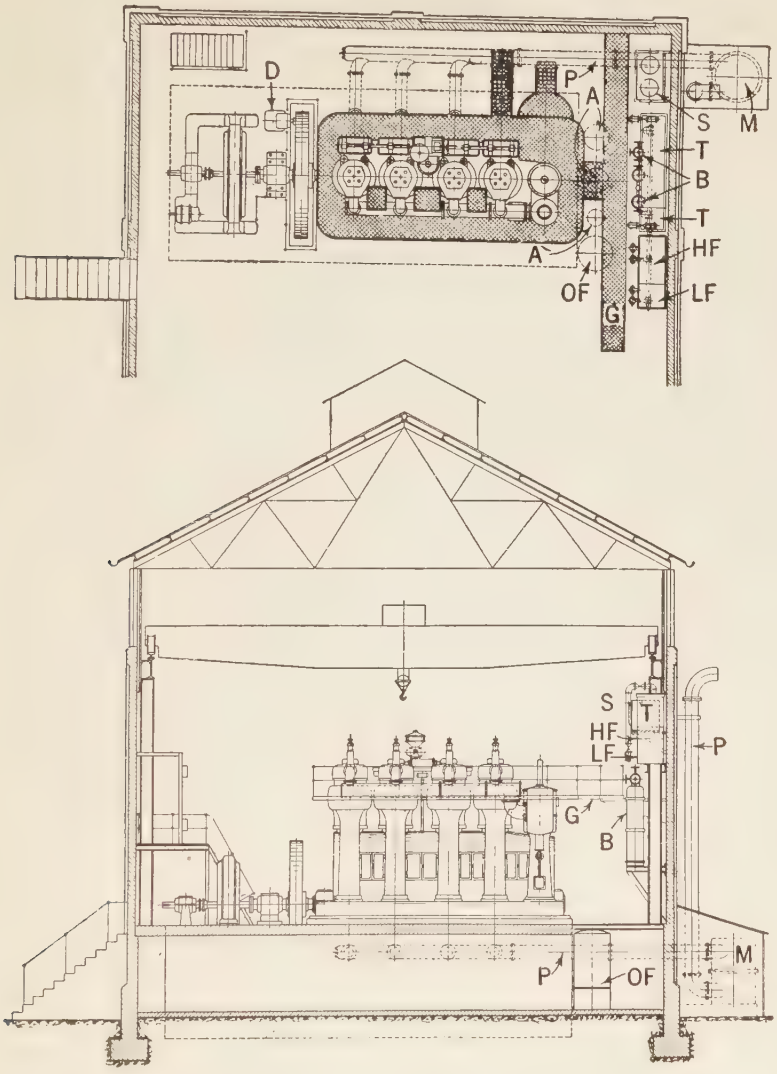


Fig. 13.—Plan and elevation of a Nordberg Diesel engine plant.

interior view of an internal-combustion engine plant in which Diesel engines are used to drive an air compressor and electric generators is shown by Fig. 14.

By the elimination of or by the reduction of certain thermal losses, internal-combustion-engine plants are often able to show higher thermal

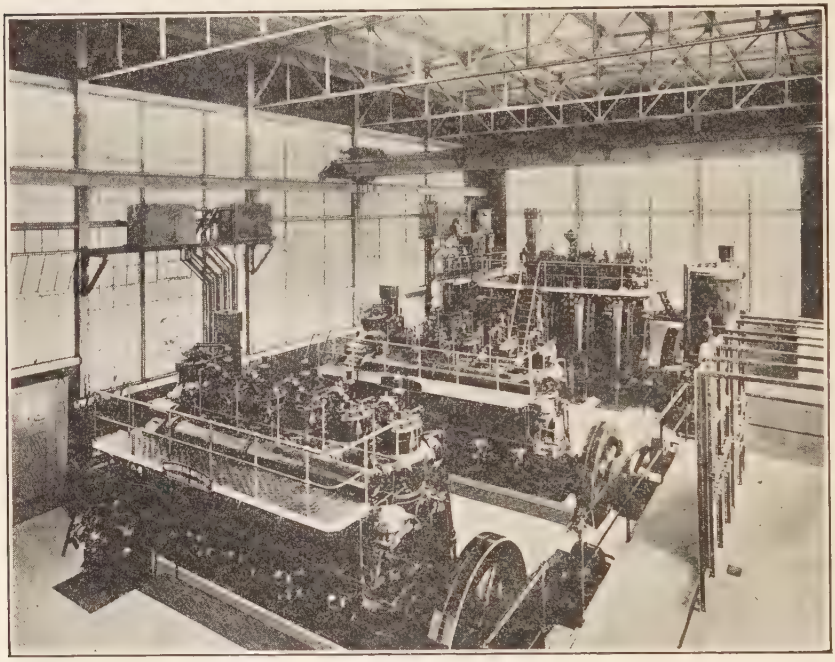


FIG. 14.—Interior of a Diesel engine plant.

efficiencies than are possible with other heat engines. Plants using internal-combustion engines may be expected to convert from 10 to 30 per cent of the heat energy of the fuel supplied into useful output.

CHAPTER IV

STEAM AND STEAM CALORIMETRY

28. Steam Generation.—A sufficiently rapid transfer of heat to water held in an open or closed vessel produces changes in the physical state of the water. The effects are: increased vibration of the molecules, a rise in the temperature, and a decrease in density of the liquid. All these changes continue until the temperature at which boiling occurs is reached. At the temperature of ebullition, bubbles of gas form within the water and rise to its surface where they burst. The rapidity with which the bubbles are formed is dependent upon the rate of heat transfer to the water and its movement due to convection currents. The conversion from a liquid into a gaseous state produces a vapor, known as steam, which exists as a gas but does not follow entirely the laws of a perfect gas. Water, when completely vaporized, is as clear and colorless as clean dry air.

The temperature at which boiling or vaporization occurs is controlled by the absolute pressure exerted upon the water. The boiling temperature is definitely fixed for each pressure and is less at lower pressures than it is at higher pressures. Water held in an open vessel at an atmospheric pressure of 14.696 lb. per sq. in. abs. will boil rapidly at a temperature of 212 deg. F. If the vessel is closed and the pressure increased to 300 lb. per sq. in. abs., the temperature at which vaporization occurs is 417.33 deg. F. A reduction of the absolute pressure to one inch of mercury within the closed vessel will result in the water boiling at a temperature of 79.06 deg. F. The vaporization temperatures of water under various absolute pressures are shown by the steam temperature curve of Fig. 15.

29. Saturated Steam.—Steam which exists at the temperature corresponding to its absolute pressure is defined as *saturated*. Saturated steam may be completely free from unvaporized water particles or it may carry water globules in suspension. Thus, saturated steam may be in either a dry or a wet condition. The term saturated does not indicate a moisture content in the steam, but does, however, indicate that the steam temperature is the normal one corresponding to the absolute pressure.

In the production of *saturated* steam the total heat added to each

pound of water is utilized in two stages. The first stage raises the temperature of the water from that of its initial condition to that which corresponds to the boiling point at the existing absolute pressure.

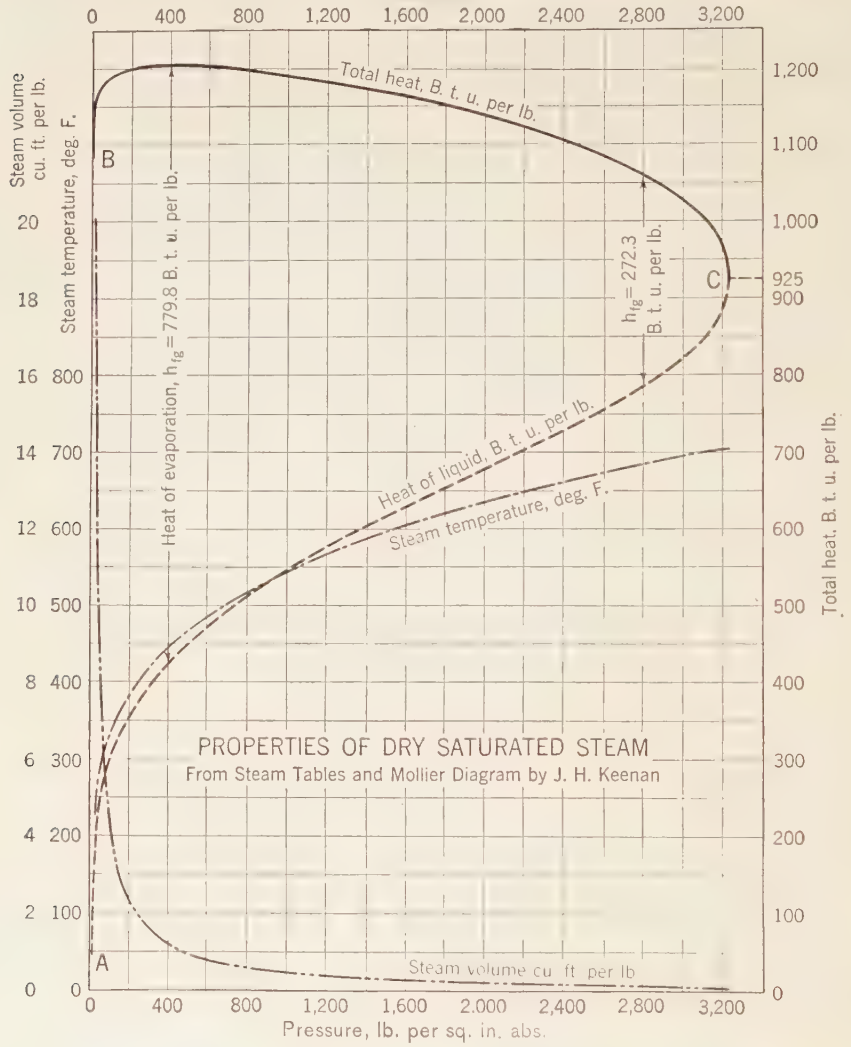

Steam volume cu. ft. per lb.
Steam temperature, deg. F.
Total heat, B. t. u. per lb.
Heat of evaporation, h_{fg} = 779.8 B. t. u. per lb.
h_{fg} = 272.3 B. t. u. per lb.
Heat of liquid, B. t. u. per lb.
Steam temperature, deg. F.
Steam volume cu ft per lb
Total heat, B. t. u. per lb.
PROPERTIES OF DRY SATURATED STEAM
From Steam Tables and Mollier Diagram by J. H. Keenan
A
B
C
Pressure, lb. per sq. in. abs.
0 400 800 1,200 1,600 2,000 2,400 2,800 3,200
0 2 4 6 8 10 12 14 16 18 20
0 100 200 300 400 500 600 700 800
1,000 1,100 1,200
925 900

Fig. 15.—Temperature, heat, and volume of dry saturated steam at various absolute pressures.

The heat necessary to accomplish this is the *heat of the liquid*. This stage is accomplished by a comparatively small increase of volume. The second stage in the process of steam formation is the addition of an amount of heat necessary to convert the liquid at its boiling point

into a vapor. The latter quantity is the *heat of evaporation* or *latent heat of vaporization*. The vaporization takes place at a constant temperature corresponding to the boiling point.

"The Properties of Dry and Saturated Steam" as compiled by Professor J. H. Keenan are given in abridged form by Tables II and III. The values shown are reproduced by permission and are based on research data secured by the General Electric Company and the American Society of Mechanical Engineers. Figure 15 is a graphical representation of a part of the data of Table II. The steam table data are based on a weight of one pound of water or steam, and the total heats and entropy changes are reckoned above a base temperature of 32 deg. F.

30. Heat of Liquid.—The amount of heat in B.t.u. necessary to raise one pound of water from a temperature of 32 deg. F. to the temperature of boiling, at any given pressure, is *the heat of the liquid.* The heat of the liquid per pound of water may be computed as

$$h_f = c_{pw}(t - 32), \tag{47}$$

where h_f = heat of the liquid, B.t.u. per lb.
c_{pw} = mean specific heat of the liquid.
t = vaporization temperature, deg. F.

The amounts of heat energy added to the water, as heat of the liquid, increase as the pressure and temperature rise. The rate of the increase in the amount of heat added is not uniform for equal increments of pressure rise, but is variable as indicated by the heat of liquid curve AC of Fig. 15. The mean specific heat of water is not constant but varies with different temperature ranges. Heat of the liquid is used to increase the temperature of the water and to effect, at the expense of energy, small volume changes in the liquid. Consequently, the shapes of the steam temperature and the heat of the liquid curves are not alike as is shown by Fig. 15. The amount of the variation is small at low temperatures so that, for temperatures less than 212 deg. F., c_{pw} may be taken with small error as equal to 1.0. For temperatures greater than 212 deg. F. the most accurate results are obtained by use of the heats of liquid given in Tables II and III.

31. Heat of Evaporation.—The heat in B.t.u. necessary to convert one pound of liquid into dry steam at the same temperature and pressure is *the total heat of evaporation* or *the latent heat of vaporization*, h_{fg}. The numerical quantities for the total latent heat of vaporization are given under the heading of "Total Heat of Evaporation" in Tables II and III. As the absolute pressure of steam increases, the heat of evapora-

TABLE II

Saturated Steam: Pressure Table

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
½" Hg	58.83	0.01603	1256.9	1256.9	26.88	1058.8	1085.7	0.0533	2.0422	2.0955	½" Hg
¾" Hg	70.44	0.01605	856.5	856.5	38.47	1052.5	1091.0	0.0754	1.9856	2.0609	¾" Hg
1" Hg	79.06	0.01607	652.7	652.7	47.06	1047.8	1094.9	0.0914	1.9451	2.0365	1" Hg
1½" Hg	91.75	0.01610	445.3	445.3	59.72	1040.8	1100.6	0.1147	1.8877	2.0024	1½" Hg
2" Hg	101.17	0.01613	339.5	339.5	69.10	1035.7	1104.8	0.1316	1.8468	1.9784	2" Hg
2½" Hg	108.73	0.01616	275.2	275.2	76.63	1031.5	1108.1	0.1450	1.8148	1.9598	2½" Hg
3" Hg	115.08	0.01618	231.8	231.8	82.96	1027.9	1110.8	0.1561	1.7885	1.9446	3" Hg
1.0	101.76	0.01614	333.8	333.9	69.69	1035.3	1105.0	0.1326	1.8442	1.9769	1.0
2.0	126.10	0.01623	173.94	173.96	93.97	1021.6	1115.6	0.1750	1.7442	1.9192	2.0
3.0	141.49	0.01630	118.84	118.86	109.33	1012.7	1122.0	0.2009	1.6847	1.8856	3.0
4.0	152.99	0.01636	90.72	90.74	120.83	1005.9	1126.8	0.2198	1.6420	1.8618	4.0
5.0	162.25	0.01641	73.59	73.61	130.10	1000.4	1130.6	0.2348	1.6088	1.8435	5.0
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1133.7	0.2473	1.5814	1.8287	6.0
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1136.4	0.2580	1.5582	1.8162	7.0
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1138.9	0.2674	1.5379	1.8053	8.0
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1141.0	0.2758	1.5200	1.7958	9.0
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1143.0	0.2834	1.5040	1.7874	10.0
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1144.8	0.2903	1.4894	1.7797	11.0
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1146.4	0.2968	1.4760	1.7727	12.0
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1147.9	0.3027	1.4636	1.7663	13.0
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1149.3	0.3082	1.4521	1.7604	14.0
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	14.696
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1151.8	0.3184	1.4312	1.7496	16.0
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1154.0	0.3274	1.4127	1.7402	18.0
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1156.0	0.3356	1.3960	1.7317	20.0
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1157.8	0.3431	1.3809	1.7240	22.0
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1159.5	0.3500	1.3670	1.7170	24.0
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1161.0	0.3564	1.3542	1.7106	26.0
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1162.4	0.3624	1.3422	1.7046	28.0
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1163.7	0.3680	1.3310	1.6990	30.0
32.0	254.05	0.01701	12.923	12.940	222.50	942.5	1165.0	0.3732	1.3206	1.6938	32.0
34.0	257.58	0.01704	12.209	12.226	226.09	940.0	1166.1	0.3783	1.3107	1.6890	34.0
36.0	260.94	0.01707	11.570	11.587	229.51	937.7	1167.2	0.3830	1.3014	1.6844	36.0
38.0	264.16	0.01710	10.998	11.015	232.79	935.5	1168.3	0.3876	1.2925	1.6800	38.0
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1169.2	0.3919	1.2840	1.6759	40.0
42.0	270.21	0.01715	10.010	10.027	238.95	931.2	1170.2	0.3961	1.2759	1.6720	42.0
44.0	273.06	0.01717	9.582	9.599	241.86	929.2	1171.1	0.4000	1.2682	1.6683	44.0
46.0	275.81	0.01719	9.189	9.207	244.67	927.2	1171.9	0.4039	1.2608	1.6647	46.0
48.0	278.45	0.01722	8.829	8.846	247.37	925.4	1172.7	0.4076	1.2537	1.6613	48.0
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1173.5	0.4111	1.2469	1.6580	50.0
52.0	283.49	0.01726	8.189	8.206	252.52	921.7	1174.3	0.4145	1.2404	1.6549	52.0
54.0	285.90	0.01728	7.902	7.919	254.99	920.0	1175.0	0.4178	1.2340	1.6518	54.0
56.0	288.23	0.01730	7.636	7.653	257.38	918.3	1175.7	0.4210	1.2279	1.6489	56.0
58.0	290.50	0.01732	7.388	7.405	259.71	916.6	1176.4	0.4241	1.2220	1.6461	58.0
60.0	292.71	0.01735	7.155	7.172	261.98	915.0	1177.0	0.4271	1.2162	1.6434	60.0
62.0	294.85	0.01737	6.937	6.955	264.18	913.4	1177.6	0.4300	1.2107	1.6407	62.0
64.0	296.94	0.01739	6.732	6.749	266.33	911.9	1178.2	0.4329	1.2053	1.6382	64.0
66.0	298.98	0.01741	6.539	6.556	268.43	910.4	1178.8	0.4356	1.2001	1.6357	66.0
68.0	300.98	0.01743	6.357	6.375	270.49	908.9	1179.4	0.4384	1.1950	1.6333	68.0
70.0	302.92	0.01744	6.186	6.203	272.49	907.4	1179.9	0.4410	1.1900	1.6310	70.0
72.0	304.82	0.01746	6.024	6.041	274.45	906.0	1180.5	0.4435	1.1852	1.6287	72.0
74.0	306.68	0.01748	5.870	5.887	276.37	904.6	1181.0	0.4460	1.1805	1.6265	74.0
76.0	308.50	0.01750	5.723	5.741	278.25	903.2	1181.5	0.4485	1.1759	1.6244	76.0
78.0	310.28	0.01752	5.584	5.602	280.09	901.9	1182.0	0.4509	1.1714	1.6223	78.0
80.0	312.03	0.01754	5.452	5.470	281.90	900.5	1182.4	0.4532	1.1670	1.6202	80.0
82.0	313.74	0.01756	5.325	5.343	283.67	899.2	1182.9	0.4555	1.1627	1.6182	82.0
84.0	315.42	0.01757	5.204	5.222	285.42	897.9	1183.4	0.4578	1.1586	1.6163	84.0
86.0	317.06	0.01759	5.089	5.107	287.13	896.7	1183.8	0.4599	1.1545	1.6144	86.0
88.0	318.68	0.01761	4.979	4.997	288.80	895.4	1184.2	0.4621	1.1505	1.6126	88.0
90.0	320.27	0.01763	4.874	4.892	290.45	894.2	1184.6	0.4642	1.1465	1.6107	90.0
92.0	321.83	0.01764	4.773	4.791	292.07	893.0	1185.0	0.4663	1.1427	1.6090	92.0
94.0	323.37	0.01766	4.676	4.694	293.67	891.8	1185.4	0.4683	1.1389	1.6072	94.0
96.0	324.88	0.01768	4.584	4.602	295.25	890.6	1185.8	0.4703	1.1352	1.6055	96.0
98.0	326.37	0.01769	4.494	4.512	296.80	889.4	1186.2	0.4723	1.1316	1.6038	98.0

TABLE II—*Continued*

SATURATED STEAM: PRESSURE TABLE

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1186.6	0.4742	1.1280	1.6022	100.0
102.0	329.27	0.01773	4.326	4.344	299.83	887.1	1186.9	0.4761	1.1245	1.6006	102.0
104.0	330.68	0.01774	4.247	4.265	301.30	886.0	1187.3	0.4779	1.1211	1.5990	104.0
106.0	332.08	0.01776	4.171	4.189	302.76	884.9	1187.6	0.4798	1.1177	1.5974	106.0
108.0	333.44	0.01777	4.097	4.115	304.19	883.8	1188.0	0.4816	1.1144	1.5959	108.0
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1188.3	0.4834	1.1111	1.5944	110.0
112.0	336.12	0.01780	3.958	3.976	307.00	881.6	1188.6	0.4851	1.1079	1.5930	112.0
114.0	337.43	0.01782	3.892	3.910	308.36	880.6	1188.9	0.4868	1.1048	1.5915	114.0
116.0	338.72	0.01783	3.828	3.846	309.71	879.5	1189.2	0.4885	1.1017	1.5901	116.0
118.0	340.01	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	118.0
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1189.8	0.4918	1.0956	1.5874	120.0
122.0	342.50	0.01788	3.652	3.670	313.67	876.4	1190.1	0.4934	1.0926	1.5860	122.0
124.0	343.73	0.01789	3.597	3.615	314.96	875.4	1190.4	0.4950	1.0897	1.5847	124.0
126.0	344.94	0.01791	3.542	3.560	316.23	874.4	1190.6	0.4965	1.0868	1.5834	126.0
128.0	346.14	0.01792	3.487	3.505	317.49	873.4	1190.9	0.4981	1.0840	1.5821	128.0
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1191.2	0.4996	1.0812	1.5808	130.0
132.0	348.48	0.01795	3.383	3.401	319.95	871.5	1191.4	0.5011	1.0784	1.5796	132.0
134.0	349.64	0.01796	3.335	3.353	321.17	870.5	1191.7	0.5026	1.0757	1.5783	134.0
136.0	350.78	0.01798	3.288	3.306	322.37	869.6	1191.9	0.5041	1.0730	1.5771	136.0
138.0	351.91	0.01799	3.242	3.260	323.56	868.6	1192.2	0.5056	1.0703	1.5759	138.0
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1192.4	0.5070	1.0677	1.5747	140.0
142.0	354.14	0.01802	3.155	3.173	325.91	866.7	1192.6	0.5084	1.0651	1.5735	142.0
144.0	355.22	0.01804	3.112	3.130	327.06	865.8	1192.9	0.5098	1.0625	1.5724	144.0
146.0	356.31	0.01805	3.071	3.089	328.20	864.9	1193.1	0.5112	1.0600	1.5712	146.0
148.0	357.37	0.01806	3.031	3.049	329.32	864.0	1193.3	0.5126	1.0575	1.5701	148.0
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1193.5	0.5140	1.0550	1.5690	150.0
152.0	359.47	0.01809	2.954	2.972	331.54	862.2	1193.7	0.5153	1.0526	1.5679	152.0
154.0	360.51	0.01810	2.917	2.935	332.64	861.3	1193.9	0.5166	1.0502	1.5668	154.0
156.0	361.53	0.01812	2.882	2.900	333.72	860.4	1194.1	0.5180	1.0478	1.5658	156.0
158.0	362.54	0.01813	2.846	2.864	334.80	859.5	1194.3	0.5193	1.0454	1.5647	158.0
160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1194.5	0.5205	1.0431	1.5636	160.0
162.0	364.54	0.01816	2.779	2.797	336.91	857.8	1194.7	0.5218	1.0408	1.5626	162.0
164.0	365.52	0.01817	2.746	2.764	337.95	857.0	1194.9	0.5230	1.0385	1.5616	164.0
166.0	366.50	0.01818	2.715	2.733	338.99	856.1	1195.1	0.5243	1.0363	1.5606	166.0
168.0	367.46	0.01819	2.683	2.701	340.01	855.2	1195.3	0.5255	1.0340	1.5596	168.0
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1195.4	0.5268	1.0318	1.5586	170.0
172.0	369.37	0.01822	2.623	2.641	342.04	853.6	1195.6	0.5280	1.0296	1.5576	172.0
174.0	370.31	0.01823	2.594	2.612	343.04	852.7	1195.8	0.5292	1.0275	1.5566	174.0
176.0	371.24	0.01825	2.566	2.584	344.03	851.9	1196.0	0.5304	1.0253	1.5557	176.0
178.0	372.16	0.01826	2.538	2.556	345.01	851.1	1196.1	0.5315	1.0232	1.5548	178.0
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1196.3	0.5327	1.0211	1.5538	180.0
182.0	374.00	0.01828	2.484	2.502	346.97	849.5	1196.4	0.5339	1.0190	1.5529	182.0
184.0	374.90	0.01829	2.458	2.476	347.94	848.6	1196.6	0.5350	1.0169	1.5520	184.0
186.0	375.78	0.01831	2.433	2.451	348.89	847.9	1196.8	0.5362	1.0149	1.5511	186.0
188.0	376.67	0.01832	2.407	2.425	349.83	847.1	1196.9	0.5373	1.0129	1.5502	188.0
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1197.0	0.5384	1.0109	1.5493	190.0
192.0	378.42	0.01834	2.359	2.377	351.70	845.5	1197.2	0.5395	1.0089	1.5484	192.0
194.0	379.27	0.01835	2.335	2.353	352.61	844.7	1197.3	0.5406	1.0070	1.5475	194.0
196.0	380.13	0.01837	2.312	2.330	353.53	844.0	1197.5	0.5417	1.0050	1.5467	196.0
198.0	380.97	0.01838	2.289	2.307	354.43	843.2	1197.6	0.5427	1.0031	1.5458	198.0
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1197.8	0.5438	1.0012	1.5450	200.0
205.0	383.89	0.01842	2.213	2.231	357.56	840.5	1198.1	0.5465	0.9964	1.5429	205.0
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1198.4	0.5491	0.9918	1.5409	210.0
215.0	387.93	0.01847	2.113	2.131	361.91	836.8	1198.7	0.5516	0.9873	1.5389	215.0
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1199.0	0.5540	0.9829	1.5369	220.0
225.0	391.81	0.01853	2.0208	2.0393	366.10	833.2	1199.3	0.5565	0.9786	1.5350	225.0
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1199.6	0.5588	0.9743	1.5332	230.0
235.0	395.56	0.01859	1.9367	1.9553	370.15	829.7	1199.8	0.5612	0.9702	1.5313	235.0
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1200.1	0.5635	0.9661	1.5295	240.0
245.0	399.20	0.01864	1.8589	1.8775	374.09	826.2	1200.3	0.5658	0.9620	1.5278	245.0

TABLE II—*Continued*

SATURATED STEAM: PRESSURE TABLE

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
250.0	400.97	0.01867	1.8223	1.8410	376.02	824.5	1200.5	0.5680	0.9581	1.5261	250.0
260.0	404.43	0.01872	1.7536	1.7723	379.78	821.2	1201.0	0.5723	0.9504	1.5227	260.0
270.0	407.79	0.01877	1.6895	1.7083	383.44	818.0	1201.4	0.5765	0.9430	1.5194	270.0
280.0	411.06	0.01882	1.6302	1.6490	387.02	814.7	1201.8	0.5805	0.9357	1.5163	280.0
290.0	414.24	0.01887	1.5745	1.5934	390.50	811.6	1202.1	0.5845	0.9287	1.5132	290.0
300.0	417.33	0.01892	1.5225	1.5414	393.90	808.5	1202.4	0.5883	0.9220	1.5102	300.0
320.0	423.29	0.01901	1.4279	1.4469	400.47	802.5	1203.0	0.5957	0.9089	1.5046	320.0
340.0	428.96	0.01910	1.3439	1.3630	406.75	796.6	1203.4	0.6027	0.8965	1.4992	340.0
360.0	434.39	0.01918	1.2689	1.2881	412.80	790.9	1203.7	0.6094	0.8846	1.4940	360.0
380.0	439.59	0.01927	1.2015	1.2208	418.61	785.3	1203.9	0.6157	0.8733	1.4891	380.0
400.0	444.58	0.0194	1.1407	1.1601	424.2	779.8	1204.1	0.6218	0.8625	1.4843	400.0
420.0	449.38	0.0194	1.0853	1.1047	429.6	774.5	1204.1	0.6277	0.8520	1.4798	420.0
440.0	454.01	0.0195	1.0345	1.0540	434.8	769.3	1204.1	0.6334	0.8420	1.4753	440.0
460.0	458.48	0.0196	0.9881	1.0077	439.9	764.1	1204.0	0.6388	0.8322	1.4711	460.0
480.0	462.80	0.0197	0.9456	0.9633	444.9	759.0	1203.9	0.6441	0.8228	1.4670	480.0
500.0	466.99	0.0198	0.9063	0.9261	449.7	754.0	1203.7	0.6493	0.8137	1.4630	500.0
520.0	471.05	0.0198	0.8701	0.8899	454.4	749.0	1203.5	0.6543	0.8048	1.4591	520.0
540.0	474.99	0.0199	0.8363	0.8562	459.0	744.1	1203.2	0.6592	0.7962	1.4554	540.0
560.0	478.82	0.0200	0.8047	0.8247	463.6	739.3	1202.9	0.6639	0.7878	1.4517	560.0
580.0	482.55	0.0201	0.7751	0.7952	468.0	734.5	1202.5	0.6686	0.7796	1.4482	580.0
600.0	486.17	0.0202	0.7475	0.7677	472.3	729.8	1202.1	0.6731	0.7716	1.4447	600.0
620.0	489.71	0.0202	0.7217	0.7419	476.6	725.1	1201.7	0.6775	0.7638	1.4413	620.0
640.0	493.16	0.0203	0.6972	0.7175	480.8	720.5	1201.2	0.6818	0.7562	1.4380	640.0
660.0	496.53	0.0204	0.6744	0.6948	484.9	715.9	1200.8	0.6861	0.7487	1.4348	660.0
680.0	499.82	0.0205	0.6527	0.6732	488.9	711.3	1200.2	0.6902	0.7414	1.4316	680.0
700.0	503.04	0.0206	0.6321	0.6527	492.9	706.8	1199.7	0.6943	0.7342	1.4285	700.0
720.0	506.19	0.0206	0.6128	0.6334	496.8	702.4	1199.2	0.6983	0.7272	1.4255	720.0
740.0	509.28	0.0207	0.5944	0.6151	500.6	697.9	1198.6	0.7022	0.7203	1.4225	740.0
760.0	512.30	0.0208	0.5769	0.5977	504.4	693.5	1198.0	0.7060	0.7136	1.4196	760.0
780.0	515.27	0.0209	0.5602	0.5811	508.2	689.2	1197.4	0.7098	0.7069	1.4167	780.0
800.0	518.18	0.0209	0.5444	0.5653	511.8	684.9	1196.7	0.7135	0.7004	1.4139	800.0
820.0	521.03	0.0210	0.5293	0.5503	515.5	680.6	1196.0	0.7171	0.6940	1.4111	820.0
840.0	523.83	0.0211	0.5149	0.5360	519.0	676.4	1195.4	0.7207	0.6877	1.4084	840.0
860.0	526.58	0.0212	0.5013	0.5225	522.6	672.1	1194.7	0.7242	0.6815	1.4057	860.0
880.0	529.29	0.0213	0.4881	0.5094	526.0	667.9	1194.0	0.7277	0.6754	1.4031	880.0
900.0	531.95	0.0213	0.4756	0.4969	529.5	663.8	1193.3	0.7311	0.6694	1.4005	900.0
920.0	534.56	0.0214	0.4635	0.4849	532.9	659.7	1192.6	0.7344	0.6635	1.3980	920.0
940.0	537.13	0.0215	0.4520	0.4735	536.2	655.6	1191.8	0.7377	0.6577	1.3954	940.0
960.0	539.66	0.0216	0.4409	0.4625	539.6	651.5	1191.1	0.7410	0.6520	1.3930	960.0
980.0	542.14	0.0217	0.4303	0.4520	542.8	647.5	1190.3	0.7442	0.6464	1.3905	980.0
1000.0	544.58	0.0217	0.4202	0.4419	546.0	643.5	1189.6	0.7473	0.6408	1.3881	1000.0
1050.0	550.53	0.0219	0.3960	0.4179	554.0	633.6	1187.6	0.7550	0.6273	1.3822	1050.0
1100.0	556.28	0.0222	0.3738	0.3960	561.7	623.9	1185.6	0.7624	0.6141	1.3765	1100.0
1150.0	561.81	0.0224	0.3540	0.3764	569.2	614.3	1183.5	0.7695	0.6014	1.3709	1150.0
1200.0	567.14	0.0226	0.3356	0.3582	576.5	604.9	1181.4	0.7764	0.5891	1.3656	1200.0
1250.0	572.30	0.0228	0.3187	0.3415	583.6	595.6	1179.2	0.7831	0.5772	1.3603	1250.0
1300.0	577.32	0.0230	0.3029	0.3259	590.6	586.3	1177.0	0.7897	0.5654	1.3552	1300.0
1350.0	582.21	0.0232	0.2884	0.3116	597.5	577.2	1174.7	0.7962	0.5540	1.3501	1350.0
1400.0	586.96	0.0235	0.2748	0.2983	604.3	568.1	1172.4	0.8024	0.5428	1.3452	1400.0
1450.0	591.58	0.0237	0.2621	0.2858	611.0	559.1	1170.0	0.8086	0.5318	1.3404	1450.0
1500.0	596.08	0.0239	0.2502	0.2741	617.5	550.2	1167.6	0.8146	0.5212	1.3357	1500.0
1600.0	604.74	0.0244	0.2284	0.2528	630.2	532.6	1162.7	0.8262	0.5003	1.3265	1600.0
1700.0	612.98	0.0249	0.2089	0.2338	642.5	515.0	1157.5	0.8373	0.4801	1.3174	1700.0
1800.0	620.86	0.0254	0.1913	0.2167	654.7	497.2	1151.8	0.8482	0.4601	1.3083	1800.0
1900.0	628.39	0.0260	0.1754	0.2014	666.8	478.9	1145.7	0.8589	0.4402	1.2990	1900.0
2000.0	635.6	0.0265	0.1610	0.1875	679.0	460.0	1139.0	0.8696	0.4200	1.2896	2000.0
2200.0	649.2	0.0277	0.1346	0.1623	703.7	420.0	1123.8	0.8912	0.3788	1.2700	2200.0
2400.0	661.9	0.0292	0.1112	0.1404	729.4	376.4	1105.8	0.9133	0.3356	1.2488	2400.0
2600.0	673.8	0.0310	0.0895	0.1205	756.7	327.8	1084.5	0.9364	0.2892	1.2257	2600.0
2800.0	684.9	0.0333	0.0688	0.1021	786.7	272.3	1058.9	0.9618	0.2379	1.1996	2800.0
3000.0	695.2	0.0367	0.0477	0.0844	823.1	202.5	1025.6	0.9922	0.1754	1.1676	3000.0
3200.0	704.9	0.0459	0.0142	0.0601	887.0	75.9	962.9	1.0461	0.0651	1.1112	3200.0
3226.0	706.1	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	3226.0

TABLE III

Saturated Steam: Temperature Table

Temp. Fahr.	Abs. Press.	Specific Volume			Total Heat			Entropy			Temp. Fahr.
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
t	p	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
32°	0.0887	0.01602	3301	3301	0.00	1073.4	1073.4	0.0000	2.1834	2.1834	32°
34	0.0961	0.01602	3059	3059	2.01	1072.3	1074.3	0.0041	2.1723	2.1764	34
36	0.1041	0.01602	2835	2835	4.03	1071.2	1075.3	0.0081	2.1614	2.1695	36
38	0.1126	0.01602	2632	2632	6.04	1070.2	1076.2	0.0122	2.1505	2.1627	38
40°	0.1217	0.01602	2445	2445	8.05	1069.1	1077.1	0.0162	2.1397	2.1559	40°
42	0.1315	0.01602	2272	2272	10.05	1068.0	1078.0	0.0202	2.1290	2.1492	42
44	0.1420	0.01602	2112	2112	12.05	1066.9	1078.9	0.0242	2.1184	2.1426	44
46	0.1532	0.01602	1965.3	1965.3	14.06	1065.8	1079.8	0.0282	2.1078	2.1360	46
48	0.1652	0.01602	1829.7	1829.7	16.06	1064.7	1080.8	0.0321	2.0974	2.1295	48
50°	0.1780	0.01602	1704.8	1704.8	18.06	1063.6	1081.7	0.0361	2.0870	2.1231	50°
52	0.1918	0.01602	1588.3	1588.3	20.06	1062.5	1082.6	0.0400	2.0767	2.1167	52
54	0.2063	0.01602	1482.4	1482.4	22.06	1061.4	1083.5	0.0439	2.0665	2.1104	54
56	0.2219	0.01603	1383.5	1383.5	24.05	1060.3	1084.4	0.0478	2.0564	2.1042	56
58	0.2384	0.01603	1292.8	1292.8	26.05	1059.3	1085.3	0.0517	2.0464	2.0980	58
60°	0.2561	0.01603	1208.0	1208.0	28.05	1058.2	1086.2	0.0555	2.0364	2.0919	60°
62	0.2749	0.01603	1129.7	1129.7	30.05	1057.1	1087.1	0.0594	2.0265	2.0859	62
64	0.2949	0.01604	1057.1	1057.1	32.04	1056.0	1088.0	0.0632	2.0167	2.0799	64
66	0.3162	0.01604	989.7	989.6	34.04	1054.9	1089.0	0.0670	2.0069	2.0739	66
68	0.3388	0.01605	927.1	927.1	36.03	1053.8	1089.8	0.0708	1.9973	2.0680	68
70°	0.3628	0.01605	869.0	869.0	38.03	1052.7	1090.8	0.0746	1.9877	2.0622	70°
72	0.3883	0.01606	814.9	815.0	40.02	1051.6	1091.7	0.0783	1.9782	2.0565	72
74	0.4153	0.01606	764.8	764.8	42.02	1050.6	1092.6	0.0820	1.9687	2.0507	74
76	0.4440	0.01606	718.0	718.0	44.01	1049.5	1093.5	0.0858	1.9593	2.0451	76
78	0.4744	0.01607	674.5	674.5	46.00	1048.4	1094.4	0.0895	1.9500	2.0395	78
80°	0.5067	0.01607	633.8	633.8	48.00	1047.3	1095.3	0.0932	1.9407	2.0340	80°
82	0.5409	0.01608	595.9	595.9	50.00	1046.2	1096.2	0.0969	1.9316	2.0285	82
84	0.5772	0.01608	560.5	560.5	52.00	1045.1	1097.1	0.1006	1.9224	2.0230	84
86	0.6153	0.01609	527.6	527.7	54.00	1044.0	1098.0	0.1042	1.9134	2.0176	86
88	0.6555	0.01609	497.1	497.1	56.00	1042.9	1098.9	0.1079	1.9044	2.0123	88
90°	0.6980	0.01610	468.5	468.5	58.00	1041.8	1099.8	0.1115	1.8955	2.0070	90°
92	0.7429	0.01610	441.7	441.8	59.98	1040.7	1100.7	0.1152	1.8866	2.0018	92
94	0.7902	0.01611	416.8	416.8	61.97	1039.6	1101.6	0.1188	1.8778	1.9966	94
96	0.8403	0.01612	393.3	393.3	63.96	1038.5	1102.5	0.1224	1.8691	1.9915	96
98	0.8930	0.01612	371.4	371.4	65.94	1037.4	1103.4	0.1260	1.8604	1.9864	98
100°	0.9487	0.01613	350.8	350.8	67.93	1036.3	1104.2	0.1295	1.8518	1.9813	100°
105	1.1009	0.01615	305.0	305.0	72.91	1033.5	1106.4	0.1384	1.8305	1.9689	105
110	1.274	0.01616	265.8	265.8	77.89	1030.8	1108.6	0.1472	1.8095	1.9567	110
115	1.470	0.01618	232.3	232.3	82.89	1027.9	1110.8	0.1559	1.7889	1.9448	115
120°	1.692	0.01620	203.5	203.6	87.88	1025.1	1113.0	0.1646	1.7685	1.9331	120°
125	1.941	0.01622	178.9	178.9	92.87	1022.2	1115.1	0.1731	1.7485	1.9216	125
130	2.221	0.01625	157.62	157.64	97.86	1019.4	1117.2	0.1816	1.7288	1.9104	130
135	2.536	0.01627	139.15	139.17	102.85	1016.5	1119.3	0.1901	1.7094	1.8995	135
140	2.887	0.01629	123.20	123.22	107.84	1013.6	1121.4	0.1984	1.6903	1.8887	140
145°	3.280	0.01632	109.29	109.31	112.84	1010.6	1123.5	0.2067	1.6715	1.8782	145°
150	3.716	0.01634	97.21	97.23	117.84	1007.7	1125.5	0.2149	1.6530	1.8679	150
155	4.201	0.01637	86.64	86.66	122.85	1004.7	1127.6	0.2231	1.6347	1.8578	155
160	4.739	0.01639	77.38	77.40	127.85	1001.8	1129.6	0.2312	1.6168	1.8479	160
165	5.334	0.01642	69.26	69.28	132.85	998.8	1131.7	0.2392	1.5991	1.8383	165
170°	5.990	0.01645	62.13	62.14	137.85	995.8	1133.7	0.2472	1.5816	1.8288	170°
175	6.716	0.01648	55.81	55.82	142.86	992.8	1135.7	0.2551	1.5644	1.8195	175
180	7.510	0.01650	50.26	50.28	147.87	989.8	1137.7	0.2629	1.5475	1.8105	180
185	8.382	0.01654	45.35	45.36	152.87	986.8	1139.7	0.2707	1.5308	1.8016	185
190	9.336	0.01656	40.99	41.01	157.89	983.8	1141.7	0.2785	1.5144	1.7929	190
195°	10.385	0.01660	37.10	37.12	162.91	980.8	1143.7	0.2862	1.4982	1.7844	195°
200	11.525	0.01663	33.65	33.67	167.94	977.7	1145.6	0.2938	1.4822	1.7760	200
205	12.769	0.01665	30.57	30.59	172.97	974.6	1147.6	0.3014	1.4664	1.7678	205
210	14.123	0.01669	27.82	27.83	177.99	971.5	1149.5	0.3089	1.4508	1.7597	210
212	14.696	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	212
215°	15.591	0.01673	25.35	25.37	183.02	968.3	1151.3	0.3164	1.4353	1.7517	215°
220	17.188	0.01676	23.14	23.16	188.06	965.1	1153.1	0.3238	1.4200	1.7439	220
225	18.915	0.01680	21.15	21.17	193.09	961.8	1154.9	0.3313	1.4049	1.7362	225
230	20.78	0.01683	19.371	19.388	198.15	958.6	1156.7	0.3386	1.3900	1.7286	230
235	22.80	0.01687	17.761	17.778	203.21	955.3	1158.5	0.3460	1.3752	1.7212	235

TABLE III—*Continued*

SATURATED STEAM: TEMPERATURE TABLE

Temp. Fahr.	Abs. Press. Lb./Sq. In.	Specific Volume			Total Heat			Entropy			Temp. Fahr.
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
t	p	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
240°	24.97	0.01690	16.307	16.324	208.26	952.0	1160.2	0.3532	1.3607	1.7138	240°
245	27.31	0.01695	14.991	15.008	213.33	948.6	1161.9	0.3604	1.3462	1.7066	245
250	29.82	0.01698	13.807	13.824	218.39	945.2	1163.6	0.3675	1.3320	1.6995	250
255	32.53	0.01702	12.726	12.743	223.47	941.8	1165.3	0.3746	1.3179	1.6925	255
260	35.43	0.01706	11.745	11.762	228.55	938.4	1166.9	0.3817	1.3040	1.6856	260
265°	38.54	0.01710	10.854	10.871	233.65	934.9	1168.6	0.3888	1.2901	1.6789	265°
270	41.85	0.01714	10.044	10.061	238.74	931.4	1170.1	0.3958	1.2765	1.6723	270
275	45.40	0.01719	9.304	9.321	243.84	927.8	1171.6	0.4028	1.2630	1.6658	275
280	49.20	0.01723	8.626	8.644	248.95	924.2	1173.2	0.4097	1.2496	1.6593	280
285	53.25	0.01728	8.007	8.024	254.08	920.6	1174.7	0.4166	1.2363	1.6529	285
290°	57.55	0.01732	7.442	7.459	259.20	917.0	1176.2	0.4234	1.2233	1.6467	290°
295	62.13	0.01737	6.923	6.940	264.34	913.4	1177.7	0.4303	1.2103	1.6406	295
300	67.01	0.01742	6.446	6.464	269.48	909.6	1179.1	0.4370	1.1974	1.6345	300
305	72.18	0.01747	6.009	6.026	274.64	905.8	1180.4	0.4438	1.1847	1.6285	305
310	77.68	0.01752	5.606	5.623	279.80	902.1	1181.9	0.4505	1.1721	1.6226	310
315	83.50	0.01757	5.234	5.252	284.98	898.2	1183.2	0.4571	1.1596	1.6167	315
320°	89.65	0.01762	4.892	4.910	290.17	894.4	1184.6	0.4638	1.1472	1.6110	320°
325	96.16	0.01768	4.573	4.591	295.38	890.5	1185.9	0.4705	1.1349	1.6054	325
330	103.03	0.01773	4.285	4.303	300.59	886.5	1187.1	0.4770	1.1227	1.5998	330
335	110.31	0.01779	4.016	4.034	305.81	882.5	1188.3	0.4836	1.1106	1.5942	335
340	117.99	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	340
345°	126.10	0.01791	3.535	3.553	316.30	874.3	1190.6	0.4966	1.0866	1.5832	345°
350	134.62	0.01797	3.321	3.338	321.55	870.2	1191.8	0.5031	1.0748	1.5779	350
355	143.58	0.01804	3.121	3.139	326.82	866.0	1192.8	0.5095	1.0630	1.5725	355
360	153.01	0.01810	2.936	2.954	332.10	861.7	1193.8	0.5160	1.0514	1.5673	360
365	162.93	0.01817	2.764	2.782	337.40	857.4	1194.8	0.5224	1.0397	1.5621	365
370°	173.33	0.01823	2.604	2.622	342.71	853.0	1195.7	0.5288	1.0282	1.5570	370°
375	184.23	0.01830	2.455	2.473	348.05	848.6	1196.7	0.5352	1.0167	1.5519	375
380	195.70	0.01836	2.315	2.333	353.39	844.1	1197.5	0.5415	1.0053	1.5468	380
385	207.71	0.01844	2.185	2.203	358.76	839.5	1198.3	0.5479	0.9939	1.5418	385
390	220.29	0.01850	2.0631	2.0816	364.14	834.9	1199.0	0.5542	0.9826	1.5368	390
395	233.47	0.01858	1.9489	1.9675	369.54	830.3	1199.8	0.5605	0.9714	1.5319	395
400°	247.25	0.01865	1.8421	1.8608	374.96	825.5	1200.4	0.5668	0.9602	1.5270	400°
410	276.72	0.01880	1.6493	1.6681	385.86	815.8	1201.6	0.5792	0.9381	1.5173	410
420	308.82	0.01896	1.4792	1.4982	396.84	805.8	1202.7	0.5916	0.9161	1.5077	420
430	343.71	0.01911	1.3295	1.3486	407.91	795.5	1203.4	0.6039	0.8942	1.4982	430
440	381.59	0.01928	1.1965	1.2158	419.07	784.9	1203.9	0.6162	0.8724	1.4887	440
450°	422.61	0.0195	1.0782	1.0977	430.3	773.8	1204.1	0.6284	0.8507	1.4792	450°
460	466.94	0.0196	0.9730	0.9927	441.7	762.3	1204.0	0.6407	0.8290	1.4696	460
470	514.76	0.0198	0.8793	0.8991	453.2	750.3	1203.5	0.6530	0.8071	1.4601	470
480	566.26	0.0200	0.7951	0.8151	465.0	737.8	1202.8	0.6654	0.7852	1.4506	480
490	621.67	0.0202	0.7195	0.7398	477.0	724.7	1201.7	0.6779	0.7632	1.4410	490
500°	681.09	0.0205	0.6516	0.6721	489.1	711.1	1200.2	0.6904	0.7410	1.4314	500°
510	744.74	0.0207	0.5903	0.6110	501.6	696.9	1198.4	0.7031	0.7187	1.4218	510
520	812.72	0.0210	0.5347	0.5557	514.2	682.1	1196.3	0.7158	0.6963	1.4121	520
530	885.31	0.0213	0.4845	0.5058	527.0	666.8	1193.8	0.7286	0.6738	1.4024	530
540	962.73	0.0216	0.4394	0.4610	540.0	651.0	1191.0	0.7414	0.6512	1.3926	540
550°	1045.4	0.0219	0.3982	0.4201	553.2	634.5	1187.8	0.7543	0.6285	1.3828	550°
560	1133.4	0.0223	0.3605	0.3828	566.7	617.5	1184.2	0.7672	0.6056	1.3728	560
570	1227.6	0.0227	0.3261	0.3488	580.4	599.7	1180.2	0.7802	0.5825	1.3626	570
580	1327.2	0.0231	0.2949	0.3180	594.4	581.3	1175.7	0.7932	0.5592	1.3524	580
590	1432.7	0.0236	0.2664	0.2900	608.7	562.2	1170.8	0.8064	0.5356	1.3420	590
600°	1544.6	0.0241	0.2401	0.2642	623.2	542.3	1165.5	0.8198	0.5118	1.3316	600°
610	1663.2	0.0247	0.2159	0.2406	638.0	521.4	1159.5	0.8332	0.4875	1.3208	610
620	1788.8	0.0254	0.1933	0.2186	653.4	499.2	1152.5	0.8470	0.4623	1.3093	620
630	1921.9	0.0261	0.1721	0.1982	669.5	474.8	1144.3	0.8612	0.4358	1.2970	630
640	2062.8	0.0269	0.1522	0.1791	686.6	447.9	1134.5	0.8763	0.4073	1.2836	640
650°	2211.4	0.0278	0.1331	0.1610	705.2	417.7	1122.8	0.8924	0.3764	1.2688	650°
660	2368.6	0.0290	0.1148	0.1437	725.3	383.6	1108.8	0.9097	0.3426	1.2523	660
670	2534.2	0.0304	0.0966	0.1269	747.5	344.4	1091.9	0.9287	0.3049	1.2336	670
680	2709.7	0.0322	0.0781	0.1102	772.6	298.5	1071.2	0.9499	0.2619	1.2119	680
690	2896.8	0.0347	0.0589	0.0936	803.0	241.2	1044.2	0.9755	0.2098	1.1852	690
700°	3096.4	0.0394	0.0353	0.0747	846.3	157.0	1003.2	1.0117	0.1354	1.1471	700°
706.1	3226.0	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	706.1

tion decreases and becomes zero at the critical pressure of 3226 lb. per sq. in. abs. At the critical pressure, water passes directly into steam without the addition of latent heat, and the total heat of dry saturated steam under those conditions is equal to the heat of the liquid, 925 B.t.u. per lb. The intercepts between curves *AC* and *BC* of Fig. 15 show graphically the decrease in the latent heat of vaporization, or heat of evaporation, as the steam pressure is made greater.

The total heat of evaporation is made up of two parts, the *external* and the *internal* latent heats. The external heat of evaporation is the heat equivalent of the external work necessary to change the volume of one pound of water to the volume occupied by one pound of the dry steam formed at the existing pressure. The external work equivalent in B.t.u. is expressed as

$$APv_{fg} = \frac{144p(v_g - v_f)}{778}. \tag{48}$$

APv_{fg} = external heat of evaporation, B.t.u. per lb.
p = steam pressure, lb. per sq. in. abs.
v_g = volume of steam formed, cu. ft. per lb.
v_f = volume of water at vaporization temperature, cu. ft. per lb.
v_{fg} = volume increase at vaporization, cu. ft. per lb.

The internal heat of evaporation ρ is the heat equivalent, in B.t.u., of the internal energy necessary to overcome the molecular attraction of the water molecules at the vaporization temperature. The internal heat of evaporation is equal to the total heat of evaporation h_{fg} minus the external heat of evaporation APv_{fg}.

$$\rho = h_{fg} - APv_{fg}. \tag{49}$$

32. Total Heat of Dry Saturated Steam.—The total heat or enthalpy of one pound of dry saturated steam is the sum of the heat of the liquid and the heat of evaporation:

$$h_g = h_f + h_{fg}. \tag{50}$$

h_g = total heat above 32 deg. F., B.t.u. per lb.
h_f = heat of the liquid, B.t.u. per lb.
h_{fg} = heat of evaporation, B.t.u. per lb.

The total heat of dry saturated steam is dependent upon its pressure as shown by the numerical quantities given under the heading of "Total Heat of Vapor, h_g," in Tables II and III. Curve *BC*, "Total Heat of Dry Saturated Steam," Fig. 15, indicates the variation and

shows that the total heat energy is a maximum of 1204.1 B.t.u. per lb. at the absolute pressures of 400 to 450 lb. per sq. in. Beyond the latter pressure the total heat of the dry vapor decreases to the minimum of 925 B.t.u. per lb. at the critical pressure of 3226 lb. sq. in. abs.

When use is made of equation 50, the values of h_f and h_{fg} are taken from either Table II or III for the absolute steam pressure corresponding to that of the steam under consideration. Tables II and III are not complete in that the "Properties of Dry Saturated Steam" are not given for all pressures. Examination of Fig. 15 will indicate that curves showing the properties of steam are not straight-line relations. However, when the properties are not given in the tables for various desired pressures, numerical data may be obtained by interpolating between the data given for the pressures nearest to the pressure to be used. Such a procedure will give data which are very nearly correct if the pressures used are not too widely separated.

33. Specific Volume and Density of Dry Saturated Steam.—The specific volumes v_g, in cubic feet, occupied by one pound of dry saturated steam are given in the fifth column of Tables II and III. The volume of one pound of steam varies inversely as the pressure, as is shown by Fig. 15. The volumes per pound of steam are very much greater at low pressures than they are at higher pressures.

The density d of dry saturated steam is the weight of steam in pounds per cubic foot. The density is the reciprocal of the specific volume v_g per pound, and is

$$d = \frac{1}{v_g} . \tag{51}$$

The density varies as the pressure, thus the higher the pressure the greater the weight of the steam per cubic foot. The volumes v_f per pound of the liquid or the water are given in the third column of the Steam Tables, and the density may be computed by use of equation 51.

34. Quality of Steam.—Saturated steam may or may not carry entrained water. Moisture present in steam may be due to several causes. Steam conveyed by piping may suffer a reduction in heat content through unavoidable or other losses, so that a part of the latent heat or heat of evaporation of the steam is given up and water particles are formed in the steam. The generation of the steam may have been imperfect on account of poor design of the boiler, high rate of evaporation maintained, or poor boiler feedwater.

The dry steam content per pound of wet steam is the *quality* of the steam. The quality x of steam may be expressed either as a percentage, or as the weight of dry steam existing within a pound of wet steam.

35. Total Heat of Wet Saturated Steam.—The total heat or enthalpy of a pound of wet saturated steam is less than that of a pound of dry saturated steam under the same pressure conditions. When the water is not completely vaporized, the heat carried as the heat of evaporation is less. The quality affects only the heat of evaporation and does not change the heat of the liquid at any given pressures. The quality factor must be applied only to the heat of evaporation, and not to the heat of the liquid or the total heat of a pound of dry saturated steam. The total heat of a pound of wet steam is

$$h_g = h_f + xh_{fg}. \tag{52}$$

h_g = total heat of wet steam, B.t.u. per lb.
x = quality of the steam.
h_f = heat of the liquid, B.t.u. per lb.
h_{fg} = heat of evaporation, B.t.u. per lb.

36. Superheated Steam.—The temperature and total heat content of saturated steam, at any pressure, may be increased by the application of additional heat. When the steam temperature is above saturation temperature, corresponding to the steam pressure, the steam is superheated. The superheating process usually takes place in a separate piece of equipment where the steam is not in contact with its liquid.

The total heat content of a pound of superheated steam is made up of three items: (1) the heat of the liquid; (2) the heat of evaporation; and (3) the heat added as superheat. In equation form the foregoing is expressed as

$$h = h_f + h_{fg} + c_p(t_{sh} - t). \tag{53}$$

h = total heat of superheated steam, B.t.u. per lb.
h_f = heat of liquid, B.t.u. per lb.
h_{fg} = heat of evaporation, B.t.u. per lb.
c_p = mean specific heat of superheated steam at the temperature and pressure of superheat.
t_{sh} = temperature of the superheated steam, deg. F.
t = temperature of saturated steam at the existing pressure, deg. F.

The specific heat of superheated steam varies with the existent pressure and the amount of the superheating. The temperature difference $(t_{sh} - t)$ is the number of degrees of superheat, and is indicative of the rise of the temperature above that of saturation conditions. Unless the mean specific heat of the superheated steam is known, the heat added as superheat cannot be computed from the temperature rise of the steam. Table IV gives values of the mean specific heats of

TABLE IV

MEAN SPECIFIC HEATS OF SUPERHEATED STEAM

Steam Press., Lb. per Sq. In. Abs.	Sat. Temp., Deg. F.	Actual Steam Temperature, Deg. F.											
		200	250	300	350	400	450	500	600	700	800	900	1000
10	193.21	0.486	0.469	0.468	0.466	0.466	0.466	0.467	0.469	0.472	0.476	0.480	0.485
15	213.03	.....	0.484	0.478	0.475	0.472	0.472	0.472	0.473	0.476	0.480	0.484	0.488
20	227.96	.....	0.500	0.488	0.482	0.478	0.476	0.476	0.477	0.479	0.483	0.486	0.491
50	281.01	.....	.....	0.547	0.523	0.510	0.503	0.499	0.495	0.494	0.496	0.499	0.502
100	327.83	.....	.....	.....	0.590	0.558	0.542	0.530	0.519	0.514	0.514	0.514	0.516
150	358.43	...	..	....	....	0.609	0.580	0.562	0.542	0.533	0.530	0.528	0.526
200	381.82	...	..	..	.....	0.659	0 619	0.593	0.564	0.551	0.542	0.540	0.539
250	400.97	.....	...	...	..	...	0 661	0.625	0 586	0.568	0.560	0.554	0.550
300	417.33	..	..	..	....	....	0 700	0.656	0.607	0.585	0.573	0.565	0.560
400	444.58	..	..	...	..	....	0 778	0.726	0.652	0.619	0.601	0.588	0.589
500	466.99	.....	.....	.....	.....	.....	.....	0.812	0.703	0.658	0.630	0.613	0.602
600	486.17	..	...	..	....	..	..	0 912	0.761	0.695	0.660	0.638	0.624
700	503.04	...	...	.	...	..	...	.....	0.824	0.735	0.691	0.664	0.646
800	518.18	.....	.....	.....	.....	.....	.....	.....	0.897	0.778	0.723	0.690	0.668
1000	544 58	....	.	...	.	.	...	....	1.067	0 870	0 788	0.742	0.714
1200	567.14	....	...	.	..	.	...	..	1 275	0.971	0.858	0.798	0.761
1400	586.96	..	..	..	..	.	..	....	1 538	1.087	0.931	0.858	0.810
1600	604.74	...	....	..	....	....	...	....	...	1 217	1.010	0.919	0.861
1800	620.86	....	..	..	....	..	..	....	...	1 375	1.098	0.988	0.915

superheated steam computed from the "Properties of Superheated Steam," Table V. The values given in Table IV are for various absolute pressures and number of degrees Fahrenheit of steam temperature. The variations of the mean specific heats of superheated steam, with different amounts of superheat at several pressures, are shown graphically by the curves of Fig. 16. The "Properties of Superheated Steam" for various pressures and steam temperatures, selected from Professor Keenan's Steam Tables and printed by permission, are given in Table V. By the use of Table V the total heat of superheated steam may be obtained directly without the calculations indicated by equation 53.

The advantages of superheating steam are: (1) the additional heat imparted to the vapor causes it to behave more nearly as a perfect gas; (2) the temperature ranges in certain pieces of steam-using equipment may be increased; and (3) steam condensation losses may be eliminated or lessened in pipelines and steam-using engines and turbines.

37. Steam Quality Determination.—The dryness or quality of saturated steam is determined by the use of some type of a steam calorimeter. If the quality of the steam x is known, the moisture content in per cent is equal to $(100 - x)$.

TABLE V

Superheated Steam

Abs. Press. Lb./Sq. In. (Sat. Temp.)		Sat. Water	Sat. Steam	Temperature—Degrees Fahrenheit 150°	200°	250°	300°	350°	400°	450°	500°	550°	600°	700°	800°	900°	1000°
	v	0.02	333.9	362.7	392.5	422.3	452.1	481.9	511.7	541.6	571.3	601.1	630.9	690.6	750.2	809.8	869.4
1	h	69.7	1105.0	1127.0	1149.8	1172.4	1195.0	1217.8	1240.7	1263.9	1287.2	1310.9	1334.6	1383.0	1432.6	1483.3	1535.2
(101.76)	s	0.1326	1.9769	2.0144	2.0503	2.0835	2.1142	2.1433	2.1707	2.1968	2.2218	2.2458	2.2688	2.3125	2.3535	2.3922	2.4291
	v	0.02	73.61		78.17	84.19	90.21	96.21	102.19	108.19	114.16	120.14	126.11	138.05	149.99	161.91	173.83
5	h	130.1	1130.6		1148.2	1171.3	1194.2	1217.2	1240.2	1263.5	1286.9	1310.7	1334.4	1382.9	1432.5	1483.2	1535.1
(162.25)	s	0.2348	1.8435		1.8712	1.9050	1.9361	1.9654	1.9930	2.0192	2.0443	2.0683	2.0914	2.1351	2.1761	2.2149	2.2517
	v	0.02	38.45		38.88	41.92	44.98	48.00	51.01	54.02	57.02	60.02	63.01	68.99	74.96	80.92	86.89
10	h	161.1	1143.0		1146.3	1169.9	1193.2	1216.4	1239.6	1262.9	1286.4	1310.1	1334.0	1382.6	1432.3	1483.1	1535.0
(193.21)	s	0.2834	1.7874		1.7925	1.8271	1.8587	1.8838	1.9161	1.9430	1.9676	1.9916	2.0148	2.0586	2.0997	2.1385	2.1753
	v	0.02	26.82			28.40	30.52	32.59	34.65	36.70	38.75	40.79	42.83	46.91	50.97	55.03	59.09
14.696	h	180.0	1150.2			1168.6	1192.2	1215.6	1239.0	1262.4	1286.0	1309.7	1333.7	1382.4	1432.1	1482.9	1534.9
(212.00)	s	0.3119	1.7564			1.7832	1.8154	1.8452	1.8731	1.8996	1.9249	1.9490	1.9722	2.0161	2.0572	2.0961	2.1330
	v	0.02	20.10			20.79	22.36	23.90	25.43	26.94	28.45	29.95	31.46	34.46	37.44	40.43	43.42
20	h	196.1	1156.0			1167.0	1191.1	1214.8	1238.3	1261.9	1285.5	1309.3	1333.4	1382.1	1432.0	1482.8	1534.8
(227.96)	s	0.3356	1.7317			1.7474	1.7802	1.8104	1.8386	1.8652	1.8906	1.9148	1.9380	1.9819	2.0232	2.0620	2.0989
	v	0.017	10.497				11.044	11.843	12.623	13.392	14.161	14.922	15.682	17.190	18.686	20.18	21.68
40	h	235.9	1169.2				1186.5	1211.3	1235.6	1259.7	1283.7	1307.8	1332.0	1381.2	1431.3	1482.3	1534.4
(267.24)	s	0.3919	1.6759				1.6990	1.7309	1.7599	1.7871	1.8128	1.8373	1.8607	1.9050	1.9464	1.9854	2.0224
	v	0.017	7.172				7.260	7.821	8.353	8.878	9.398	9.911	10.423	11.435	12.436	13.439	14.440
60	h	262.0	1177.0				1181.2	1207.7	1232.8	1257.4	1281.9	1306.2	1330.7	1380.2	1430.6	1481.8	1533.9
(292.71)	s	0.4271	1.6434				1.6488	1.6827	1.7128	1.7406	1.7667	1.7915	1.8151	1.8597	1.9014	1.9406	1.9775
	v	0.018	5.470					5.806	6.217	6.618	7.015	7.406	7.793	8.558	9.313	10.067	10.817
80	h	281.9	1182.4					1203.8	1229.9	1255.2	1280.0	1304.7	1329.3	1379.2	1429.9	1481.3	1533.5
(312.03)	s	0.4532	1.6202					1.6473	1.6785	1.7070	1.7336	1.7586	1.7824	1.8274	1.8694	1.9086	1.9456
	v	0.018	4.426					4.594	4.934	5.263	5.585	5.903	6.215	6.831	7.439	8.044	8.644
100	h	298.3	1186.6					1199.7	1226.9	1252.8	1278.0	1303.0	1327.9	1378.2	1429.2	1480.7	1533.1
(327.83)	s	0.4742	1.6022					1.6185	1.6512	1.6805	1.7075	1.7329	1.7569	1.8023	1.8445	1.8838	1.9209
	v	0.018	3.725					3.781	4.077	4.359	4.632	4.899	5.162	5.680	6.189	6.693	7.196
120	h	312.4	1189.8					1195.2	1223.8	1250.4	1276.1	1301.4	1326.5	1377.2	1428.5	1480.2	1532.7
(341.26)	s	0.4918	1.5874					1.5940	1.6283	1.6584	1.6859	1.7115	1.7359	1.7816	1.8240	1.8635	1.9007
	v	0.018	3.216						3.465	3.713	3.951	4.182	4.410	4.857	5.297	5.728	6.161
140	h	324.7	1192.4						1220.5	1247.9	1274.1	1299.7	1325.1	1376.2	1427.8	1479.7	1532.2
(353.03)	s	0.5070	1.5747						1.6084	1.6393	1.6674	1.6933	1.7179	1.7640	1.8066	1.8462	1.8836
	v	0.018	2.830						3.005	3.227	3.440	3.645	3.846	4.240	4.627	5.006	5.385
160	h	335.9	1194.5						1217.1	1245.4	1272.1	1298.0	1323.7	1375.2	1427.1	1479.1	1531.8
(363.55)	s	0.5205	1.5636						1.5906	1.6224	1.6510	1.6774	1.7022	1.7487	1.7915	1.8313	1.8687
	v	0.018	2.529						2.646	2.849	3.041	3.226	3.407	3.760	4.105	4.444	4.782
180	h	346.0	1196.3						1213.5	1242.7	1270.0	1296.3	1322.2	1374.2	1426.3	1478.5	1531.4
(373.08)	s	0.5327	1.5538						1.5742	1.6073	1.6364	1.6631	1.6882	1.7351	1.7782	1.8180	1.8555
	v	0.018	2.285						2.358	2.547	2.722	2.892	3.056	3.376	3.688	3.995	4.299
200	h	355.3	1197.8						1209.8	1240.0	1267.9	1294.6	1320.8	1373.1	1425.6	1478.0	1531.0
(381.82)	s	0.5438	1.5450						1.5592	1.5934	1.6231	1.6502	1.6756	1.7228	1.7662	1.8062	1.8438
	v	0.019	2.084						2.122	2.299	2.462	2.617	2.769	3.062	3.347	3.628	3.903
220	h	364.0	1199.0						1205.9	1237.3	1265.7	1292.9	1319.3	1372.1	1424.8	1477.4	1530.6
(389.89)	s	0.5540	1.5369						1.5450	1.5805	1.6109	1.6384	1.6641	1.7117	1.7553	1.7954	1.8331
	v	0.0186	1.9156						1.9250	2.092	2.244	2.389	2.529	2.800	3.063	3.321	3.574
240	h	372.1	1200.1						1201.9	1234.4	1263.5	1291.1	1317.8	1371.0	1424.1	1476.8	1530.2
(397.40)	s	0.5635	1.5295						1.5317	1.5684	1.5996	1.6275	1.6534	1.7014	1.7453	1.7856	1.8234
	v	0.0187	1.7723							1.9165	2.060	2.195	2.327	2.579	2.823	3.062	3.295
260	h	379.8	1201.0							1231.5	1261.3	1289.3	1316.3	1370.0	1423.3	1476.3	1529.7
(404.43)	s	0.5723	1.5227							1.5571	1.5890	1.6174	1.6435	1.6919	1.7360	1.7765	1.8144
	v	0.0188	1.6490							1.7655	1.9019	2.030	2.153	2.389	2.617	2.839	3.056
280	h	387.0	1201.8							1228.4	1259.0	1287.5	1314.8	1368.9	1422.5	1475.7	1529.3
(411.06)	s	0.5805	1.5163							1.5462	1.5790	1.6078	1.6343	1.6830	1.7274	1.7680	1.8060
	v	0.0189	1.5414							1.6347	1.7648	1.8863	2.002	2.224	2.438	2.646	2.849
300	h	393.9	1202.4							1225.3	1256.7	1285.6	1313.3	1367.8	1421.7	1475.1	1528.9
(417.33)	s	0.5883	1.5102							1.5359	1.5695	1.5988	1.6256	1.6747	1.7193	1.7601	1.7983
	v	0.0191	1.3245							1.3712	1.4899	1.5981	1.7003	1.8945	2.080	2.260	2.435
350	h	409.8	1203.6							1217.1	1250.7	1280.9	1309.4	1365.1	1419.8	1473.6	1527.9
(431.71)	s	0.6061	1.4966							1.5117	1.5477	1.5783	1.6059	1.6561	1.7012	1.7424	1.7809
	v	0.0194	1.1601							1.1726	1.2828	1.3817	1.4740	1.6472	1.8119	1.9704	2.125
400	h	424.2	1204.1							1208.3	1244.3	1276.0	1305.5	1362.3	1417.7	1472.1	1526.8
(444.58)	s	0.6218	1.4843							1.4892	1.5276	1.5599	1.5884	1.6396	1.6854	1.7270	1.7658

TABLE V—*Continued*

Superheated Steam

Abs. Press. Lb./Sq. In. (Sat. Temp.)		Sat. Water	Sat. Steam	Temperature—Degrees Fahrenheit 500°	550°	600°	620°	640°	660°	680°	700°	750°	800°	850°	900°	950°	1000°
	v	0.0196	1.0303	1.1204	1.2127	1.2972	1.3299	1.3616	1.3932	1.4242	1.4548	1.5299	1.6032	1.6752	1.7455	1.8148	1.8834
450	h	437.4	1204.1	1237.6	1271.0	1301.5	1313.2	1324.9	1336.5	1348.0	1359.4	1387.8	1415.7	1443.2	1470.6	1498.2	1525.8
(456.27)	s	0.6361	1.4732	1.5091	1.5430	1.5725	1.5835	1.5942	1.6046	1.6148	1.6248	1.6487	1.6714	1.6928	1.7133	1.7331	1.7524
	v	0.0198	0.9261	0.9905	1.0775	1.1558	1.1861	1.2153	1.2444	1.2727	1.3009	1.3697	1.4365	1.5018	1.5655	1.6284	1.6903
500	h	449.7	1203.7	1230.5	1265.8	1297.3	1309.4	1321.4	1333.2	1344.9	1356.6	1385.3	1413.6	1441.5	1469.1	1496.9	1524.8
(466.99)	s	0.6493	1.4630	1.4915	1.5274	1.5579	1.5692	1.5802	1.5908	1.6012	1.6113	1.6356	1.6586	1.6802	1.7009	1.7210	1.7404
	v	0.0200	0.8402	0.8823	0.9658	1.0398	1.0679	1.0953	1.1223	1.1487	1.1746	1.2381	1.2996	1.3596	1.4181	1.4756	1.5321
550	h	461.3	1203.0	1222.8	1260.3	1293.1	1305.5	1317.7	1329.8	1341.8	1353.6	1382.8	1411.5	1439.7	1467.6	1495.8	1523.8
(476.92)	s	0.6616	1.4536	1.4747	1.5126	1.5443	1.5559	1.5671	1.5780	1.5886	1.5990	1.6236	1.6468	1.6687	1.6897	1.7100	1.7296
	v	0.0202	0.7677	0.7922	0.8728	0.9431	0.9695	0.9954	1.0206	1.0452	1.0694	1.1285	1.1855	1.2411	1.2953	1.3483	1.4003
600	h	472.3	1202.1	1214.7	1254.6	1288.7	1301.5	1314.1	1326.4	1338.6	1350.6	1380.3	1409.3	1437.8	1466.1	1494.4	1522.8
(486.17)	s	0.6731	1.4447	1.4582	1.4986	1.5316	1.5436	1.5551	1.5662	1.5770	1.5874	1.6125	1.6360	1.6582	1.6794	1.6999	1.7196
	v	0.0206	0.6527		0.7251	0.7905	0.8143	0.8376	0.8602	0.8822	0.9038	0.9559	1.0063	1.0554	1.1029	1.1484	1.1929
700	h	492.9	1199.7		1242.4	1279.7	1293.3	1306.5	1319.4	1332.0	1344.5	1375.1	1404.9	1434.1	1463.0	1491.9	1520.8
(503.04)	s	0.6943	1.4285		1.4720	1.5080	1.5208	1.5329	1.5444	1.5557	1.5665	1.5923	1.6165	1.6392	1.6608	1.6817	1.7018
	v	0.0209	0.5653		0.6128	0.6750	0.6974	0.7189	0.7395	0.7596	0.7791	0.8265	0.8723	0.9157	0.9577	0.9982	1.0374
800	h	511.8	1196.7		1228.8	1270.1	1284.6	1298.6	1312.1	1325.3	1338.2	1369.7	1400.4	1430.3	1459.9	1489.3	1518.8
(518.18)	s	0.7135	1.4139		1.4462	1.4862	1.4998	1.5127	1.5248	1.5365	1.5477	1.5744	1.5992	1.6225	1.6446	1.6659	1.6864
	v	0.0213	0.4969		0.5234	0.5844	0.6057	0.6258	0.6452	0.6639	0.6821	0.7257	0.7675	0.8072	0.8451	0.8815	0.9166
900	h	529.5	1193.3		1213.6	1259.8	1275.5	1290.3	1304.5	1318.3	1331.7	1364.2	1395.8	1426.5	1456.8	1486.8	1516.8
(531.95)	s	0.7311	1.4005		1.4208	1.4656	1.4803	1.4938	1.5067	1.5188	1.5304	1.5579	1.5835	1.6074	1.6301	1.6518	1.6727
	v	0.0217	0.4419		0.4495	0.5111	0.5317	0.5509	0.5692	0.5870	0.6040	0.6449	0.6837	0.7202	0.7547	0.7877	0.8199
1000	h	546.0	1189.6		1196.5	1248.7	1265.8	1281.6	1296.6	1311.0	1324.9	1358.5	1391.0	1422.6	1453.6	1484.3	1514.8
(544.58)	s	0.7473	1.3881		1.3949	1.4455	1.4615	1.4760	1.4895	1.5022	1.5144	1.5427	1.5691	1.5936	1.6169	1.6390	1.6603
	v	0.0222	0.3960			0.4500	0.4705	4893.0	0.5071	0.5239	0.5401	0.5788	0.6152	0.6490	0.6810	0.7115	0.7408
1100	h	561.7	1185.6			1236.6	1255.4	1272.5	1288.3	1303.4	1317.9	1352.6	1386.1	1418.7	1450.4	1481.8	1512.8
(556.28)	s	0.7624	1.3765			1.4257	1.4433	1.4590	1.4733	1.4866	1.4993	1.5285	1.5557	1.5810	1.6048	1.6274	1.6491
	v	0.0226	0.3582			0.3985	0.4189	0.4373	0.4547	0.4710	0.4865	0.5233	0.5578	0.5897	0.6195	0.6478	0.6750
1200	h	576.5	1181.4			1223.4	1244.2	1262.7	1279.6	1295.5	1310.6	1346.5	1381.1	1414.7	1447.2	1479.3	1510.8
(567.14)	s	0.7765	1.3656			1.4058	1.4254	1.4423	1.4576	1.4716	1.4848	1.5151	1.5431	1.5692	1.5937	1.6168	1.6388
	v	0.0235	0.2983			0.3144	0.3361	0.3548	0.3717	0.3874	0.4021	0.4365	0.4678	0.4964	0.5229	0.5476	0.5712
1400	h	604.3	1172.4			1192.4	1218.9	1241.3	1260.8	1278.6	1295.2	1333.8	1370.8	1406.5	1440.8	1474.3	1506.9
(586.96)	s	0.8024	1.3452			1.3643	1.3891	1.4096	1.4272	1.4429	1.4574	1.4899	1.5199	1.5477	1.5736	1.5976	1.6204
	v	0.0244	0.2528				0.2706	0.2908	0.3083	0.3239	0.3384	0.3708	0.4002	0.4264	0.4503	0.4725	0.4935
1600	h	630.2	1162.7				1188.2	1216.2	1239.6	1260.1	1278.6	1320.3	1360.0	1398.1	1434.3	1469.3	1502.9
(604.74)	s	0.8262	1.3265				1.3503	1.3760	1.3971	1.4152	1.4313	1.4666	1.4986	1.5283	1.5555	1.5808	1.6042
	v	0.0254	0.2167					0.2384	0.2574	0.2736	0.2880	0.3195	0.3472	0.3716	0.3938	0.4140	0.4330
1800	h	654.7	1151.8					1186.3	1215.3	1239.4	1260.6	1306.2	1348.7	1389.5	1427.8	1464.3	1498.8
(620.85)	s	0.8482	1.3083					1.3399	1.3661	1.3874	1.4058	1.4444	1.4787	1.5105	1.5392	1.5656	1.5897
	v	0.0265	0.1875					0.1931	0.2145	0.2320	0.2468	0.2781	0.3047	0.3279	0.3486	0.3673	0.3847
2000	h	679.0	1139.0					1149.0	1186.6	1216.0	1240.7	1291.4	1337.0	1380.5	1421.1	1459.2	1494.7
(635.61)	s	0.8696	1.2896					1.2988	1.3326	1.3586	1.3802	1.4230	1.4599	1.4938	1.5242	1.5518	1.5765
	v	0.0277	0.1623						0.1760	0.1960	0.2120	0.2438	0.2696	0.2919	0.3115	0.3289	0.3451
2200	h	703.7	1123.8						1150.9	1188.7	1218.4	1275.7	1324.8	1371.3	1414.2	1454.0	1490.4
(649.3)	s	0.8912	1.2700						1.2943	1.3278	1.3536	1.4021	1.4418	1.4779	1.5102	1.5390	1.5644
	v	0.0292	0.1404							0.1637	0.1816	0.2148	0.2401	0.2618	0.2805	0.2969	0.3121
2400	h	729.4	1105.8							1155.5	1192.8	1259.1	1312.3	1361.6	1407.1	1448.7	1486.1
(661.9)	s	0.9133	1.2488							1.2928	1.3253	1.3813	1.4244	1.4628	1.4969	1.5270	1.5531
	v	0.0310	0.1205							0.1314	0.1540	0.1896	0.2151	0.2361	0.2542	0.2699	0.2842
2600	h	756.7	1084.5							1110.0	1162.5	1241.0	1299.2	1351.9	1399.8	1443.2	1481.7
(673.8)	s	0.9364	1.2257							1.2481	1.2938	1.3602	1.4074	1.4484	1.4842	1.5157	1.5426
	v	0.0333	0.1021								0.1275	0.1674	0.1934	0.2141	0.2315	0.2466	0.2603
2800	h	786.7	1058.9								1123.8	1221.3	1285.5	1341.6	1392.1	1437.6	1477.3
(684.9)	s	0.9618	1.1996								1.2559	1.3384	1.3905	1.4342	1.4720	1.5049	1.5326
	v	0.0367	0.0844								0.0983	0.1476	0.1742	0.1947	0.2118	0.2265	0.2396
3000	h	823.1	1025.6								1066.3	1199.3	1271.1	1331.0	1384.3	1431.7	1472.9
(695.3)	s	0.9922	1.1676								1.2028	1.3155	1.3737	1.4203	1.4602	1.4946	1.5233
	v	0.0459	0.0601									0.1293	0.1572	0.1777	0.1945	0.2088	0.2214
3200	h	887.0	962.9									1174.5	1255.9	1320.1	1376.1	1425.6	1468.4
(704.9)	s	1.0461	1.1112									1.2907	1.3567	1.4067	1.4487	1.4845	1.5144
	v	0.0522	0.0522									0.1271	0.1552	0.1757	0.1924	0.2067	0.2192
3226	h	925.0	925.0									1171.2	1253.8	1318.7	1375.0	1424.9	1467.8
(706.1)	s	1.0785	1.0785									1.2874	1.3545	1.4050	1.4472	1.4832	1.5133
	v											0.1042	0.1349	0.1556	0.1720	0.1860	0.1981
3500	h											1129.7	1231.2	1303.0	1363.3	1416.0	1461.6
	s											1.2484	1.3307	1.3866	1.4318	1.4699	1.5017

Two types of steam calorimeters for the determination of either the steam quality or the moisture content are in use. These are the *separating* calorimeter, and the *throttling* or *superheating* calorimeter. Combinations of these types may be employed where the steam is of low quality, and where a greater degree of accuracy is required.

When a quality or moisture determination is made, it is imperative that a representative sample of the steam be passed through the calorimeter. A typical sample of the steam can be secured only by the use of a properly designed and located sampling nozzle. The sampling nozzle should be located preferably in a vertical main as far as possible from pipe bends and fittings. The precautions are necessary to insure the withdrawal of a sample, in which the entrained moisture is not unduly increased by condensed steam flowing along the walls of the steam main. The connection between the sampling nozzle and the calorimeter should be as short as possible. Both the calorimeter and the calorimeter connections should be well insulated.

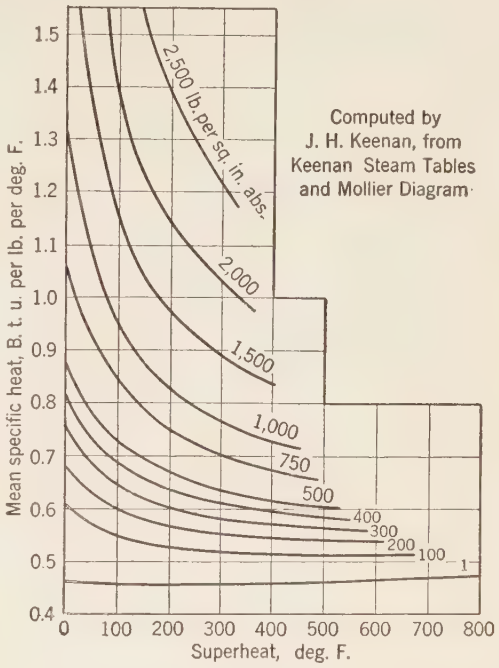


FIG. 16.—Mean specific heats of superheated steam.

38. Separating Calorimeter.—This is a device for mechanically separating the entrained moisture from the steam and collecting it so that the moisture may be accurately measured. Provision must also be made for the measurement of the amount of dried steam escaping at the calorimeter exhaust. The construction of a typical separating calorimeter is illustrated by Fig. 17.

The calorimeter body A has an inner chamber C at the top of which is placed the separating cup B. The incoming steam strikes the perforated cup and is made to reverse its direction of flow. The reversal of the steam flow, together with the impingement upon the mesh, causes the water particles to be separated and dropped to the bottom of the inner chamber. The level of the moisture collected in the inner chamber may be observed in the gage glass G connected with the chamber. A

graduated scale S is held along the gage glass; and from this scale the weight of the moisture collected within the calorimeter is read. The smallest graduation on this scale is usually 0.01 lb.; by careful estimation the weight of moisture may be read to 0.001 lb.

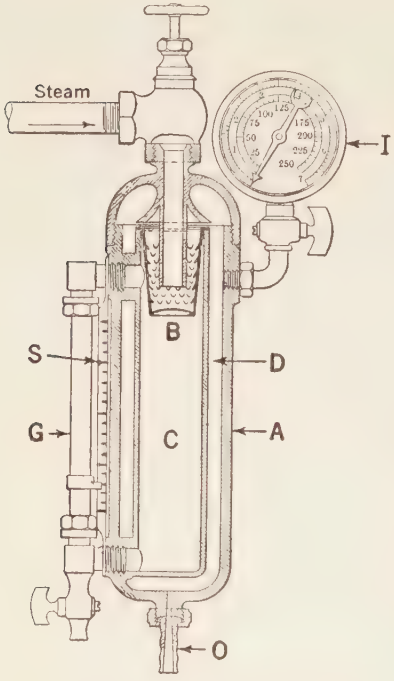


FIG. 17.—Separating calorimeter.

The dry steam escapes from the perforated baffle or cup through an annular chamber D which surrounds the inner chamber. The outer chamber serves as a steam jacket which reduces the radiation and other heat losses from the incoming steam. The steam escapes from the outer chamber through an orifice O located at the bottom of the calorimeter body. The steam may be discharged to the air or into a small condenser. The latter is preferable for the accurate determination of the weight of dry steam escaping during a given period of time, as the dry steam condensed may be weighed. A pressure gage I is attached to the outer chamber so that the pressure within the calorimeter may be determined. This gage also has graduations upon its dial indicating the flow of dry steam through the orifice, per ten minutes of time. The weight of dry steam flowing per second may also be calculated by Napier's rule when the discharge pressure is less than 0.06 of the initial pressure,

$$w = \frac{pa}{70}. \tag{54}$$

w = weight of dry steam per second, lb.
p = steam pressure, lb. per sq. in. abs.
a = area of the orifice, sq. in.

The attached pressure and flow gage is subject to many errors and may give erroneous readings. Also, the orifice may become clogged with dirt and rust, or it may be corroded so that even though the gage is mechanically correct the readings may be misleading. Likewise, the calculations of the weight of steam by Napier's rule may be unreliable, if

the diameter of the orifice is not accurately known or if the orifice is partially clogged.

The steam quality in per cent is given by the equation,

$$x = \frac{W_d \times 100}{W_d + W_m}. \tag{55}$$

x = quality of the steam, per cent.
W_d = weight of dry steam passing through the calorimeter, lb.
W_m = weight of moisture separated from W_d lb. of dry steam, lb.

Separating calorimeters should be used for the determination of the quality of low-pressure steam, or very wet steam.

39. Throttling Calorimeters. All calorimeters of this type operate upon the principle that if steam is expanded without doing any external work, or without the loss of heat, the total energy of the steam is unchanged. Radiation losses should be minimized by having the calorimeter, as well as its connection to the steam main, thoroughly insulated. The steam is expanded by being allowed to flow through a small orifice from a region of high pressure to one of low pressure.

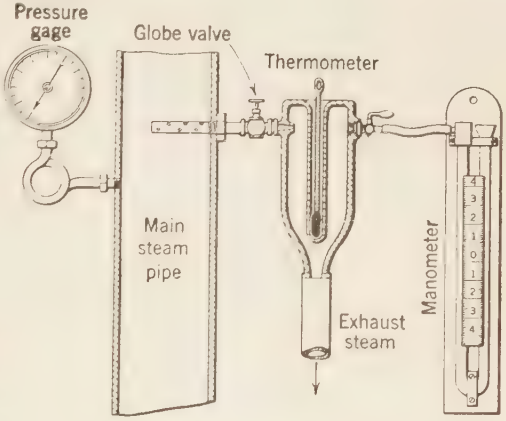


FIG. 18.—Throttling calorimeter.

One form of a throttling calorimeter is illustrated by Fig. 18. This calorimeter consists of a chamber with a naked thermometer, or a thermometer well, centrally placed within it. Steam enters the calorimeter through a small orifice located in one side of the body. A mercury manometer is attached to the calorimeter shell and is used to measure the pressure of the steam within the thermometer chamber. The calorimeter discharge may be directly to the air or through suitable piping to any convenient place. The function of the Bourdon gage is to indicate the pressure of the steam flowing to the calorimeter.

The throttling calorimeter is operated by fully opening the valve in the calorimeter connection and allowing the steam to expand through the orifice. If no work is done and there are no radiation losses, the total heat energy of the steam is unchanged during the process. However, the total heat of the high-pressure wet steam is greater than that

normal for saturated steam at the absolute pressure after expansion. This excess of heat is utilized in evaporating the moisture of the steam and superheating the steam at the lower pressure. The temperature of the low-pressure steam is measured by the thermometer inserted into the calorimeter body. The relation may be established: the total heat of the wet saturated steam before expansion, $h_{f,1} + x_1 h_{fg,1}$, is equal to the total heat of the superheated steam after expansion, or $h_{g,2} + c_p(t_c - t) = h_2$, therefore,

$$x_1 = \frac{h_{g,2} + c_p(t_c - t) - h_{f,1}}{h_{fg,1}} = \frac{h_2 - h_{f,1}}{h_{fg,1}}. \qquad (56)$$

x_1 = initial quality of the steam.
$h_{f,1}$ = heat of the liquid at the initial pressure abs., B.t.u. per lb.
$h_{fg,1}$ = heat of evaporation at the initial pressure abs., B.t.u. per lb.
$h_{g,2}$ = total heat of dry saturated steam at the calorimeter chamber pressure abs., B.t.u. per lb.
h_2 = total heat of superheated steam at the calorimeter pressure, B.t.u. per lb.
c_p = mean specific heat of superheated steam.
t_c = temperature of the superheated steam, deg. F.
t = temperature of saturated steam corresponding to the absolute pressure within the calorimeter, deg. F.

For accurate work, corrections should be properly applied to the readings of thermometers having mercury columns exposed to the surrounding air. Throttling calorimeters should be well insulated, and at least 10 deg. F. of superheat must be obtained by the expansion of the steam. Throttling calorimeters exhausting to the atmosphere may not be expected to give accurate quality determinations with steam having moisture contents greater than those listed for the following gage pressures:

Gage Pressure, Lb. per Sq. In.	Maximum Moisture Content, Per Cent.
40	2.5
150	4.5
250	5.5
600	6.0

40. Combination Calorimeters. –The improved calorimeter shown by Fig. 19 gives close approaches to the theoretical temperatures of a calorimeter. Steam enters through the sampling nozzle and is expanded in an orifice. The expanded steam enters the steam chamber and escapes at its top after traversing a U path. Momentary excess moisture

is separated and re-evaporated within the steam chamber, and the effect is indicated in direct proportion by the superheat thermometer placed in the path of the steam. A throttling, separating and re-evaporating calorimeter is formed in a single chamber, which is jacketed by the escaping steam. The jacket steam is protected against chilling by heavy insulation and a nickel-plated casing.

A steam pressure gage is necessary to indicate the initial pressure of the steam. The exhaust takes place freely to the atmosphere. The quality of the steam can be calculated by equation 56, as given for a throttling calorimeter.

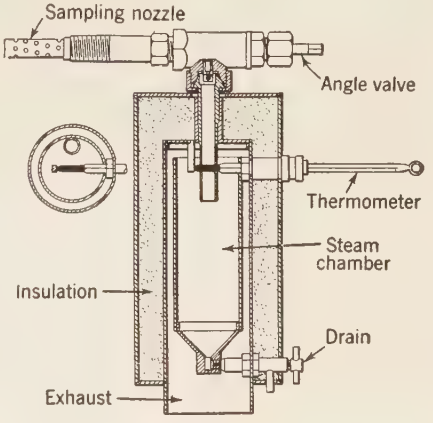


FIG. 19.—Ellison steam calorimeter.

41. Calorimeter Errors.—One of the chief errors in determinations of the quality of steam is caused by the inability to secure a composite sample of the steam. The following are some of the errors that may occur in the use of calorimeters:

Separating.—Condensation between the nozzle and the calorimeter, incomplete separation of the moisture, inaccuracies of the steam-flow gage, or outlet orifice clogged or corroded, rendering incorrect the flow gage readings or the calculation by Napier's rule.

Throttling.—Radiation losses, pressure gage errors, orifice opening clogged with dirt, thermometers in wells do not respond readily to temperature variations, and stem corrections not applied to the exposed thermometer stems.

Explicit directions for reducing calorimeter errors are given by the Instrument and Apparatus Codes of the American Society of Mechanical Engineers. Included are instructions for the proper sampling of steam in vertical and horizontal steam pipes and for the determination of the normal readings of the calorimeter.

42. Change of Entropy.—The ratio of the amount of heat added to or taken from a pound of steam to the average absolute temperature at which the addition or the abstraction is made expresses the *change of entropy*. The heat of evaporation, or latent heat of vaporization, added isothermally to one pound of water at its boiling temperature, T deg. F. abs., produces the change in entropy at vaporization h_{fg}/T or s_{fg}. Thus entropy change is the change in heat per pound per degree of absolute temperature, and is generally referred to as entropy.

The total entropy, s_g, of saturated steam is the sum of the entropy of the liquid, s_f, and the entropy of evaporation, s_{fg}. Numerical values for the entropy of one pound of dry saturated steam may be found by reference to Tables II and III.

The change of entropy of the liquid cannot be expressed so simply as the entropy of evaporation because the liquid temperature is not constant and because the specific heat of water varies with different temperatures. When heat is added to the liquid its average absolute temperature is not the numerical mean of the initial and final tempertures. Therefore, it is necessary to consider that the heat is added in infinitely small amounts dh_f at the changing absolute temperature T between the initial and the final absolute temperatures T_1 and T_2.

For any given temperature T, $ds_f = \frac{dh_f}{T}$ and the increment of heat is equal to Tds_f or $c_{pw}dT$. The summation of the infinitely small increments for a given temperature rise or the change of entropy of the liquid is

$$s_f = \int_{T_1}^{T_2} c_{pw} \frac{dT}{T}. \tag{57}$$

When the specific heat $c_{pw} = 1.0$, the expression integrated becomes

$$s_f = \log_e T_2 - \log_e T_1 = \log_e \frac{T_2}{T_1}. \tag{58}$$

Numerical data for entropy changes are given by Tables II and III, "Properties of Dry and Saturated Steam." Figure 20 is a temperature-entropy diagram for one pound of water converted into dry saturated steam at various absolute temperatures. The path of the plotted values of the entropy changes of the liquid, s_f, at various temperatures, falls along the liquid line AB, while that of total entropy of the vapor, s_g, falls along the dry and saturated line CB. Thus at a temperature of 491.6 deg. F. abs. the total entropy change, s_g, or 2.1834, is represented by the length of the horizontal line AC. In this particular case the entropy of the liquid, s_f, is zero, and the entropy at vaporization, s_{fg}, is equal to AC. In the case of water evaporated at an absolute pressure of 381.59 lb. per sq. in. or a temperature of 899.6 deg. F. abs. the entropy of the liquid, 0.6162, is shown by line ab; the entropy of evaporation, 0.8724, by line bc; and the total entropy of the vapor, 1.4887, by line ac. At an absolute temperature of 1165.7 deg. F., corresponding to the critical pressure of 3226 lb. per sq. in. abs., the entropy of the liquid, 1.0785, is equal to the total entropy, as the entropy of evaporation has become zero.

43. Entropy Changes of Wet Steam.—When steam carries moisture in suspension, the total entropy changes and those at vaporization are less than those indicated in Tables II and III. In the calculation of the heat of a pound of wet saturated steam the quality factor x affects only the heat of evaporation h_{fg}. At any given pressure the heat of the liquid is never changed by a quality factor. Hence, in the calculation of the change of entropy of a pound of water converted into wet steam, the entropy of evaporation is multiplied by the quality. The total entropy change is expressed as

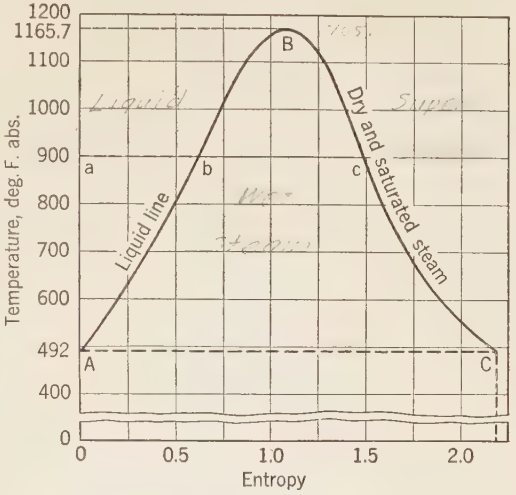


FIG. 20.—Temperature-entropy diagram for dry saturated steam.

$$s_g = s_f + x s_{fg}. \tag{59}$$

44. Entropy Changes of Superheated Steam.—The addition of a sufficient amount of heat to saturated steam without an increase in the steam pressure produces superheated steam. The steam temperature and the total entropy change are increased. Figure 21 illustrates the temperature-entropy changes of superheated steam. The change of entropy due to superheating is delineated by the curve at C sloping upward to the right. Entropy changes during superheating occur with increasing temperatures and varying specific heats. Consequently the method of calculating the changes that occur during the superheating process is similar to that employed for entropy variations of a liquid. The reader is referred to Table V, "Properties of Superheated Steam," giving the entropy changes for superheated steam.

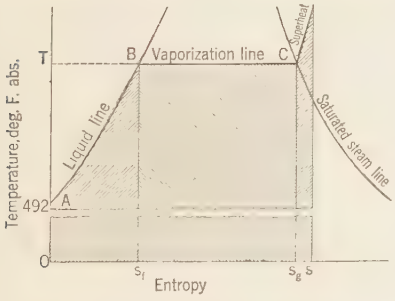


FIG. 21.—Temperature-entropy diagram, superheated steam.

45. Constant Entropy.—When adiabatic conditions are maintained the heat of a material is neither given to nor received from external bodies. Under adiabatic conditions there is no change in the total entropy. However, if steam expands adiabatically from a condition of higher temperature to one of lower temperature and work is done, there is a reduction in the heat of the steam. If the expansion of superheated steam is carried far enough, or if the steam is initially saturated, the quality of the steam after expansion will be less than 100 per cent. If the initial steam conditions and the final steam pressure after adiabatic expansion are known, the final quality may be computed. With no change of entropy during adiabatic expansion in which work is done, the initial total entropy is equal to the final total entropy.

$$s_{f,1} + x_1 s_{fg,1} = s_{f,2} + x_2 s_{fg,2}, \tag{60}$$

$$x_2 = \frac{s_{f,1} + x_1 s_{fg,1} - s_{f,2}}{s_{fg,2}}. \tag{61}$$

The numerical quantities are to be taken directly from the steam tables.

46. Total Heat-entropy Diagrams.—The Mollier diagram, Fig. 22, affords a ready and rapid aid in solving problems involving the properties of steam. From this chart the total heat and entropy changes of steam at various pressures may be read for either a dry, wet, or superheated condition. The use of the diagram permits a quick determination of the quality of steam from data given by a throttling calorimeter test. The total heat and quality of steam may be determined after adiabatic expansion, also from the chart, when the initial steam conditions are known.

Example.—Find the total heat and the total entropy of one pound of steam at an absolute pressure of 160 lb. per sq. in. and 96 per cent quality.

Solution.—Locate on the diagram, Fig. 22, the intersection of the 4 per cent moisture, or the 96 per cent quality line and the 160 lb. per sq. in. pressure line. Read along a horizontal line the total heat of the steam 1160 B.t.u. per lb. and along a vertical line the total entropy 1.522.

Example.—Steam at a pressure of 180 lb. per sq. in. abs. had 100 deg. F. of superheat. Find the total heat per pound and the total entropy.

Solution.—Find the intersection of the 180 lb. per sq. in. pressure line and the 100 deg. F. constant superheat line. The total heat corresponding to this intersection is 1255 B.t.u. per lb., and the total entropy is 1.620.

Example.—A throttling calorimeter test gave the following data: average initial steam pressure 140 lb. per sq. in. abs., calorimeter exhaust pressure 14.7 lb. per sq. in. abs., temperature of the steam in the calorimeter 260 deg. F. Find the quality of the steam.

Solution.—Locate the intersection of the 260 deg. line and the 14.7 lb. per sq. in. pressure line between the 240 and the 280 deg. steam temperature lines.

Move horizontally from this point to the 140 lb. per sq. in. pressure line and there read a moisture content of 2.2 per cent or a quality of 97.8 per cent.

Example.—Steam at 240 lb. per sq. in. abs. and 99 per cent quality is expanded adiabatically to a pressure of 3 in. of mercury abs. Find the total heat and the quality of the steam after expansion.

Solution.—From the intersection of the 240 lb. per sq. in. pressure line and the one per cent moisture line drop vertically along a line of constant entropy to the 3 in. mercury pressure line. The quality is 76.5 per cent and the total heat is 870.0 B.t.u. per lb.

PROBLEMS

1. Calculate the heat of the liquid of 25 lb. of water whose boiling temperature is 210 deg. F. Assume the mean specific heat of the water as 1.0. What absolute saturated steam pressure corresponds to the vaporization temperature?

2. Calculate the heat of the liquid of 5000 lb. of water in a steam boiler at a vaporization temperature of 395 deg. F. Mean specific heat of the water is 1.018. What is the corresponding saturated steam pressure?

3. Calculate the mean specific heats of water at the following temperatures: 175, 323, and 400 deg. F. Use the heats of the liquid as given in Table III.

4. Seven thousand pounds of water were heated from 60 deg. F. to the vaporization temperature corresponding to an absolute pressure of 177.3 lb. per sq. in. How many B.t.u. heat of the liquid were required?

5. Find the total heat of evaporation required for the vaporization of 15,000 lb. of water at boiler pressures of 187 and 615 lb. per sq. in abs.

6. Calculate the internal and the external latent heats of evaporation for one pound of dry and saturated steam at an absolute steam pressure of 425 lb. per sq. in.

7. A steam boiler delivered dry steam at a pressure of 191.5 lb. per sq. in. gage. Barometer 29.53 in. of mercury. Find: the steam temperature; the heat of the liquid; the heat of evaporation; the total heat; the volume; and the internal and the external heats of evaporation per pound.

8. Find the total heat of 35,000 lb. of dry saturated steam at 155 lb. per sq. in. gage. Barometer 28.6 in. of mercury.

9. How much heat is required to convert 65,000 lb. of water at a temperature of 190 deg. F. into dry saturated steam at a gage pressure of 200 lb. per sq. in.? Barometer 29.6 in. of mercury.

10. What is the total heat per pound of steam containing 4 per cent of moisture when it exists at the following absolute pressures: 75, 125, 275, 373, and 1200 lb. per sq. in.?

11. Find the total heat of one pound of steam of 97.5 per cent quality at the following pressures: 150, 225, 413, 648.5, and 1812 lb. per sq. in. abs.

12. How many B.t.u. must be added to 27,500 lb. of water evaporated in a steam boiler into steam of 97.3 per cent quality at an absolute pressure of 153.7 lb. per sq. in., when the temperature of the entering feedwater is 197 deg. F.? Find the volume of the steam formed.

13. A boiler has a steam space of 150 cu. ft. How many pounds of steam of 98 per cent quality are held by the boiler when the steam pressure is 173 lb. per sq. in. abs.?

14. A steam engine receiver has a volume of 4 cu. ft. How many pounds of steam of 90 per cent quality will it hold when the absolute pressure of the steam is 35 lb. per sq. in.?

15. Find the density of the liquid and the density of dry saturated steam at the following temperatures: 100, 320, 485, 657, and 700 deg. F.

16. Find the density of the liquid and the density of dry saturated steam at the following absolute pressures: 5, 25, 117, 900, and 3200 lb. per sq. in.

17. Steam at a pressure of 400 lb. per sq. in. abs. has 155.4 deg. F. of superheat. Calculate the total heat per pound and check the value given in Table V.

18. Calculate the mean specific heat of superheated steam existing at a pressure of 220 lb. per sq. in. abs. and a temperature of 500 deg. F.

19. One pound of steam at an absolute pressure of 450 lb. per sq. in. had a temperature of 750 deg. F. Compute the number of degrees of superheat, the heat of superheat, and the total heat required to generate one pound of such steam from water at 250 deg. F. How much external work was done in the formation of the steam?

20. A boiler equipped with a superheater was supplied with water at a temperature of 400 deg. F. Steam was delivered at the superheater outlet at a pressure of 625 lb. per sq. in. abs. and a total temperature of 725 deg. F. Find the heat added to 87,000 lb. of steam as superheat. Find the heat added to the feedwater to produce the steam.

21. A steam generator operated at 1450 lb. per sq. in. abs. pressure and delivered 350,000 lb. of steam per hour at a temperature of 800 deg. F. The temperature of the feedwater entering the boiler was 580 deg. F. Calculate the heat added to the feedwater per hour.

22. A steam line has dry steam escaping through an opening $\frac{1}{8}$ in. in diameter. The line pressure is 125 lb. per sq. in. abs., and the steam escapes to the atmosphere. Find the loss of steam in pounds per 24 hours.

23. A separating calorimeter operating at a steam pressure of 175 lb. per sq. in. abs. discharged 7.5 lb. of dry steam and collected 0.25 lb. of moisture in a period of 10 minutes. What was the quality of the steam? Calculate the diameter, in inches, of the calorimeter orifice.

24. Steam of 94 per cent quality was supplied to a separating calorimeter and 12 lb. of dry steam were discharged in a period of 10 minutes. How much moisture was separated from the steam per minute? What was the initial steam pressure if the calorimeter orifice diameter was 0.076 in.?

25. Wet steam was expanded in a throttling calorimeter from an initial pressure of 160 lb. per sq. in. abs. to an exhaust pressure of 14.5 lb. per sq. in. abs. The temperature of the superheated steam in the calorimeter was 284 deg. F. What was the steam quality?

26. If 10 deg. F. of superheat are the minimum allowable in steam-quality determinations by use of a throttling calorimeter, what was the maximum percentage of moisture that could have been determined under the pressure conditions of problem 25?

27. Steam of 98 per cent quality was supplied at 135 lb. per sq. in. gage pressure to a throttling calorimeter whose exhaust pressure was 0.5 lb. per sq. in. gage when the barometric pressure was 29.6 in. of mercury. What steam temperature existed in the calorimeter?

28. Calculate the change of entropy of the liquid of one pound of water between 32 and 212 deg. F.

29. Calculate the entropy change at evaporation at the following absolute steam pressures: 136, 200, 320, 600, and 1800 lb. per sq. in. Find the total entropy changes for the same pressures.

30. One pound of steam at a temperature of 500 deg. F. was expanded adiabatically from an absolute pressure of 220 lb. per sq. in. to atmospheric pressure. Barometer 29.92 in. of mercury. Calculate the steam quality after expansion, and check the result by use of the Mollier diagram.

CHAPTER V

FUELS FOR STEAM GENERATION

47. Production of Heat.—In commercial power plants the heat energy used to produce either steam or mechanical energy is released from a fuel. This heat energy is secured by the rapid oxidation of some material which may be either in its natural state or in a prepared form. The only requirements are that the fuel be anything that can be burned without great difficulty, and that sufficient quantities are available at prices not prohibitive.

48. Forms of Fuels.—The common fuels used in the production of heat for commercial purposes are in either a *solid*, a *liquid*, or a *gaseous state*. Each of these three fuel forms may be subdivided according to the method of preparation of the fuel. Listed below are the most important fuels for steam generation.

(1) Solid	Natural: coal, lignites, and wood. Prepared: coke and pulverized fuels. Refuse: wood and vegetable wastes.
(2) Liquid	Natural: crude petroleum oils. Prepared: petroleum distillates and coal tar.
(3) Gaseous	Natural: natural gas. Prepared: blast-furnace gas and coke-oven gas.

Coal in its various forms is the most commonly used fuel. Second in importance are the fuel oils. Gas may be burned alone or in conjunction with oil or pulverized coal, coke, or lignite. The past consumption of natural gas has been largely in its fields of production or adjacent territories, but the construction of long-distance, high-pressure pipelines for the transmission of natural gas has increased its usage as a steam fuel. About steel and iron plants a considerable amount of steam power is produced from blast-furnace and coke-oven gases. Wood is generally too bulky and expensive for the amount of heat contained to have an extensive usage. Manufacturing plants producing wood and certain vegetable wastes usually can burn such by-products with economical results. Fine sizes of coke and coal tar, a by-product of gas and coke manufacture, have some local usage. The chief

heat-producing constituents in any of the foregoing fuels are carbon and hydrogen or their compounds.

49. Coal.—Coal is the transformed remains of the rank and luxuriant vegetation of prehistoric ages, and is widely distributed in various localities of the world. The constituents of coal are carbon, hydrogen, oxygen, nitrogen, sulphur, ash, and moisture. Coal varies greatly in composition and characteristics which are dependent upon the changes that the coal has undergone and the impurities that have entered into it during the transformation.

The heat-producing constituents of a coal are the carbon, the hydrogen, and the sulphur. These items collectively are the burnable or combustible part of the fuel. The term *combustible* is usually applied to the part of the fuel that is represented after the moisture and ash have been deducted. Combustible as thus defined includes carbon, hydrogen, oxygen, nitrogen, and sulphur.

50. Analyses of Coal.—Coal analysis data may be obtained by the use of either the *proximate* or the *ultimate* method. Either method is accurate if properly carried out.

The *proximate analysis* is made by mechanical means and separates the coal into four constituent parts, which are by percentages, moisture, volatile matter, fixed carbon, and ash. The sum of these four percentages is equal to 100. A separate determination of the sulphur is usually made and reported along with, but independent of, the other determinations of the proximate analysis.

Moisture.—Coal contains moisture in two forms: that which is inherent in the coal and that which is due to exposure to the weather and other sources of water. All coal used in analyses and calorific determinations should be in equilibrium with the air, so that moisture is neither given to nor received from the surrounding atmosphere. Coal in such a state is said to be air dried. The loss in moisture upon air drying is secured by drying a representative sample of proper-sized coal, at a temperature of 90 deg. F., until it shows a loss of weight not to exceed 0.10 per cent per hour.

An air-dried sample is then crushed and finely ground, and further dried in an oven at a temperature of 220 deg. F. for a period of one hour. The loss of weight from this sample is then noted. From the data secured for the losses in weight, upon air drying and oven drying, the total moisture percentage of the coal is calculated.

Volatile Matter.—Coals contain hydrogen, hydrocarbons, and other gas-forming ingredients. The combination of these materials driven off by the application of heat to a dry coal sample held in a covered crucible is known as the volatile matter.

The usual procedure, in the determination of the percentage of vola-

tile matter, is to place one gram of a finely powdered, air-dried sample of coal in a small platinum crucible fitted with a tight cover. This tightly covered crucible is held in a gas or electrically heated muffle furnace at a temperature of 1700 to 1775 deg. F. for exactly 7 minutes. The crucible is then removed from the furnace and allowed to cool within a desiccator without disturbing the crucible cover. After the crucible is cold it is weighed and the loss of weight determined. When proper corrections for the moisture content of the coal have been made, the percentage of volatile matter is computed from the data secured.

Ash.—The residue of the completely burned combustible, together with the earthy matter contained in the coal, is designated as ash. The sample used in the volatile-matter determination is burned at a high temperature until it shows no further loss of weight. Care must be taken, in the burning process, that none of the lighter residue is lost from the crucible. The net weight of the residue represents the ash and earthy matter of the sample. This percentage should be adjusted to the original moisture content of the coal.

Fixed Carbon.—The uncombined carbon of the coal is designated as fixed carbon. The percentage is determined by subtracting from 100 the sum of the percentages of the moisture, volatile matter, and ash. Not all the carbon of the coal is represented by this percentage of fixed carbon, as the volatile matter contains a part of the total carbon in combination with hydrogen. Furthermore, not all the sulphur may have disappeared with the volatile matter.

Sulphur.—Sulphur is an undesirable element in a fuel because the sulphur compounds formed by oxidation may, under certain conditions, produce corrosive mixtures. Furthermore, in many cases sulphur is the cause of the fusing and clinkering of the ash. The sulphur determination is made separately by using a fused sample of the coal. The fused coal is chemically treated and the resulting precipitate collected. The precipitate formed is burned and the loss of weight noted, and from this the sulphur percentage is computed.

51. Ultimate Analysis.—When all the percentages of the various coal constituents are desired, an ultimate analysis is necessary. In such an analysis the percentages of carbon, hydrogen, nitrogen, and oxygen are determined by the use of chemical apparatus and calculations. The percentages of sulphur, moisture, and ash are secured as in the case of the proximate analysis. Ultimate analyses can be made satisfactorily only by those well trained in the work. When careful studies of combustion are necessary, ultimate analyses of the fuel are essential.

52. Reports of Fuel Analyses. The analyses of solid fuels and their heating values may be reported in three ways: (1) as received or

as fired; (2) dry or moisture free; and (3) combustible, or ash and moisture free.

Coals can best be compared with each other when placed on a dry basis, as their moisture content is variable with the weather and other conditions. The analysis of the wet coal is desirable as it represents the " as-fired condition " of the coal entering the furnace.

Wet coals may be reduced to a dry basis by dividing each of the constituent percentages, except the moisture percentage, by 1 − moisture (one minus the moisture content expressed decimally). Likewise, the percentages representing the combustible may be obtained by dividing the wet coal values, except moisture and ash, by 1 − moisture and ash (one minus the moisture and ash expressed decimally).

Example.—The proximate analysis of a coal as received gave moisture, 5.02; volatile matter, 35.67; fixed carbon, 52.26; and ash, 7.05 per cent. Express the analysis on a dry basis.

Solution.—

		Per Cent
Volatile matter	= 35.67 ÷ (1 − 0.0502)	= 37.55
Fixed carbon	= 52.26 ÷ (1 − 0.0502)	= 55.03
Ash	= 7.05 ÷ (1 − 0.0502)	= 7.42
Total		100.00

When conversion from the dry basis or combustible basis is to be made to the as-received condition, the dry or combustible values are multiplied by 1 − moisture, or by 1 − ash and moisture; and the percentages of moisture, or moisture and ash added to the respective analyses.

53. Coal Classifications.—A scheme of coal classification is difficult to make up, covering every condition encountered and sharply separating all coals into distinctive classes. Schemes of classification that are in use are based on: (1) the carbon content; (2) the ratio of the carbon to the hydrogen content; (3) the ratio between the volatile matter and the fixed carbon; (4) the percentage of volatile matter and the vein moisture content; (5) the unit heat value of the pure coal (termed unit coal) and the unit volatile matter percentage;[1] and (6) the rank of the coal.

The U. S. Geological Survey uses the term " rank " (of item 6) to designate the present stage of transition in the coal formation. The term rank signifies the completeness of the transition from the first stage, which is peat, to the final stage of almost pure graphite. The word grade is also applied in order that the quality of the rank may be defined with respect to ash, moisture, and sulphur content.

[1] "The Classification of Coal," by Samuel W. Parr; University of Illinois Engineering Experiment Station Bulletin 180, 1928.

If the Geological Survey classification is used, the progressive stages in the formation of coal are: peat, lignite, sub-bituminous, bituminous, semi-bituminous, semi-anthracite, anthracite, and almost pure graphite.

A classification of coals based upon the percentages of volatile matter and the percentages of moisture of the coal in the vein is given by Table VI. This classification is a modification of a classification made by Frazer in 1877 in a study of Pennsylvania coals.

TABLE VI

Classification of Coals by Volatile Matter and Vein Moisture

Coal	Volatile Matter, Per Cent	Vein Moisture, Per Cent
Anthracites	Below 5	
Semi-anthracites	5–10	
Semi-bituminous (Pocahontas)	15–32	
Bituminous (Eastern)	25–35	2– 4
Bituminous (Mid-Continent)	35–45	6–17
Sub-bituminous	35–45	17–20
Lignites	25–45	20–25

Peats and graphitic coals are not considered, as they are of no consequence as fuels for power purposes. Peat, when freshly dug, contains a large amount of water and cannot be burned until it has been dried. Graphitic coals can be burned only with great difficulty. In the intermediate states between peat and graphite, coal finds a wide application as a fuel. The greater part of the fuel used for power purposes falls under the classifications of the various grades of bituminous coals.

54. Physical Properties of Coals.—The physical characteristics of coals are listed in the following paragraphs.

Anthracite is a very hard coal having a shiny black luster. Anthracite coal is non-coking, and contains a high percentage of fixed carbon with a very small amount of volatile matter. This coal is slow to ignite unless the furnace temperature is high; and it requires a strong draft. Anthracite coal burns without flames or with a very short bluish flame, and is desirable where smokeless combustion is essential. The supply is very limited; thus the cost is becoming high. Fine sizes are available for use in stoker-fired installations. The principal and best field of anthracite coal in the United States is in eastern Pennsylvania.

Semi-anthracite coal occurs in small quantities in various localities of the United States and is not used commercially as a steam coal. The principal differences from a true anthracite are: it has less fixed carbon,

more volatile matter, and less luster, and it burns with longer and more luminous flames.

Semi-bituminous coal is the highest grade of bituminous coal and burns with a very small amount of smoke. This coal is softer than the anthracites, contains more volatile matter, and has a tendency to break up into small sizes during storage or transportation. The latter item is of small consequence in the generation of steam so long as the size is uniform. The principal fields of this coal in the United States are in West Virginia, Maryland, Pennsylvania, Virginia, and Tennessee.

Bituminous coals are "soft," have high percentages of volatile matter, burn with long yellow and smoky flames, and vary greatly in the percentages of volatile matter, moisture, ash, and sulphur contents. The chief distinction between the eastern bituminous coals and those of the central portion of the United States is in their percentages of volatile matter and moisture content when mined. Both high- and low-ash coals are included. A high-ash coal is usually designated as one having more than 10 per cent ash and earthy matter content; also such coals in many cases have a high-sulphur content. Bituminous coals are classified as free-burning, and caking or coking varieties. Free-burning coals do not swell up and fuse together when ignited, as do the coking or caking coals.

Sub-bituminous coals, sometimes known as black lignites, are a very low grade of bituminous coal. These coals have lost the woody structural appearance which characterizes the lignites. Sub-bituminous coals disintegrate when exposed to the air and require careful attention during storage.

Lignites are in the transition state between peat and the sub-bituminous grade of coal, and have a woody or often a clay-like appearance, together with low heat values and high moisture and high ash contents. The initial moisture content may be as high as 30 to 45 per cent, but moisture is readily lost upon exposure to the air, which may produce rapid disintegration of the fuel. Unless carefully stored, lignites are subject to spontaneous combustion. Fuels of this class are of value for local use only, but give promise of greater usefulness in a pulverized form.

55. Coal Sizes.—Coal taken from the mine during mining operations ranges in size from fine dust to large lumps. Generally the fine sizes contain more impurities and dirt, owing to segregation; consequently they do not usually have so high a heating value per pound as the larger sizes. Also, special equipment may be necessary in order to burn them. Therefore, the smaller sizes are lower in price and find a large use for the production of steam in power plants. The preparation and sizing of coal form an essential part of the industry of coal production.

The sizing operations are carried on by crushing and screening the coal to produce the various market sizes.

The sizes approved and adopted by the Anthracite Institute for the various trade names of anthracite coal are given in Table VII.

TABLE VII

Anthracite Coal Sizes

Trade Name	Diameter of Screen Openings, Inches	
	Through	Over
Broken	$4\frac{3}{8}$	$3\frac{1}{4}$
Egg	$3\frac{1}{4}$	$2\frac{7}{16}$
Stove	$2\frac{7}{16}$	$1\frac{5}{8}$
Nut	$1\frac{5}{8}$	$\frac{13}{16}$
Pea	$\frac{13}{16}$	$\frac{9}{16}$
Buck	$\frac{9}{16}$	$\frac{5}{16}$
Rice	$\frac{5}{16}$	$\frac{3}{16}$
Barley	$\frac{3}{16}$	$\frac{3}{32}$
Culm or waste	$\frac{3}{32}$	

For bituminous coals, no general classification of sizes is recognized in all localities. The sizing of bituminous coals recommended by the American Society of Mechanical Engineers employs two groups. One is for eastern and the other for western coals. The recommended sizes for the various trade names of bituminous coal sizes are given in Table VIII.

TABLE VIII

Bituminous Coal Sizes

Name	EASTERN Diameters of Screen Openings, Inches		WESTERN Diameters of Screen Openings, Inches	
	Through	Over	Through	Over
Run of mine	Unscreened coal as taken from the mine		Unscreened coal as taken from the mine	
Lump	.....	$1\frac{1}{4}$		
Lump, 6 in., 3 in., $1\frac{1}{4}$ in.	.....	.....	.....	6, 3 or $1\frac{1}{4}$
Lump, 6 in. × 3 in.	.....	.....	6	3
Lump, 3 in. × $1\frac{1}{4}$ in.	.....	.....	3	$1\frac{1}{4}$
Nut	$1\frac{1}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$
Slack	$\frac{3}{4}$			
Screenings	.....	.....	$1\frac{1}{4}$	

56. Chemical Properties of Coals.—Coals vary greatly in the percentages of their various constituents, and consequently in the heat that they will produce per pound. This variation is marked not only in the same coal field but also in the same vein and even in the same mine. Analyses of samples of the various ranks of coal, selected and adjusted from U. S. Bureau of Mines Bulletin 22, are given in Table IX. Reference should be made to the bulletin for a more complete listing of the various coals of the United States.

57. Coal Impurities.—Moisture, ash, oxygen, sulphur, and nitrogen are all impurities of coal. The effects of these impurities, either individually or collectively, have a marked influence upon the combustion of the various coals. In general, it is desirable to have a fuel which has a low content of each and all the impurities mentioned.

Nitrogen occurs in limited quantities and is chemically inert. The effect of nitrogen is small, as it acts only as a diluent of the products of combustion.

Sulphur.—Unless sulphur occurs in a free state in the coal it does not have its full heat value. In the sulphate form sulphur is worthless. Sulphur in large quantities is prone to facilitate the fusion of the ash and earthy matter, thus producing a troublesome clinker which is hard to remove from the furnace.

Ash and earthy matter represent waste materials which reduce the heat value of the fuel and which must be removed both from the fires and from the plant at a considerable expense. Ash tends to clog the air spaces in the fuel bed, reducing the rate at which fuel may be burned. Also, certain coals have an ash which fuses at low temperatures and forms clinkers. The clinkering qualities of a coal are very important from an operating standpoint. Clinkers cause deterioration of furnace linings, labor and expense in removal, as well as loss in capacity due to the fusing material sealing over the air openings in the grate.

Moisture represents a part of the coal which has no fuel value. A loss of heat energy occurs in the evaporation of moisture and in its escape with the products of combustion.

Oxygen has the general effect of lowering the heat value of a fuel by an amount equivalent to that occasioned by the same percentage of ash.

58. Coal Storage.—Users of large amounts of coal are compelled to provide facilities for storing quantities of coal for use in periods of heavy consumption or market shortages. Anthracite coal may be stored without trouble. With other coals, storage difficulties may be encountered, which are the effects of weathering upon the heating value, disintegration of the coal, and the likelihood of fire due to spontaneous combustion.

TABLE IX

Analyses of Coals of the United States

(Abstracted from U. S. Bureau of Mines Bulletin 22)

Rank	State	County	Sample	Proximate Analysis, Coal as Received				Ultimate Analysis, Coal as Received					Heat Value by Calorimeter, B.t.u. per Lb.	
				Moisture	Volatile Matter	Fixed Carbon	Ash	Sulphur	Carbon	Hydrogen	Oxygen	Nitrogen	As Rec'd	Dry
Anthracite	Colorado	Gunnison	Mine	2.70	3.32	88.15	5.83	0.80	85.38	2.98	1.19	1.12	14,099	14,490
Anthracite	Pennsylvania	Lackawanna	Car culm	5.41	7.02	71.79	15.78	0.74	72.65	2.50	2.15	0.77	12,047	12,737
Anthracite	Pennsylvania	Schuylkill	Mine	2.80	1.16	88.21	7.83	0.89	84.36	1.58	1.91	0.63	13,298	13,682
Semi-anthracite	Colorado	Gunnison	Mine	4.86	6.96	81.87	6.31	0.81	79.66	3.43	3.00	1.93	13,468	14,155
Semi-anthracite	Pennsylvania	Sullivan	Mine	3.38	8.47	76.65	11.50	0.63	78.43	3.20	1.86	1.00	13,156	13,617
Semi-bituminous	Maryland	Allegany	Car run of mine	3.42	17.65	71.84	7.09	0.84	80.17	4.34	2.69	1.45	14,162	14,665
Semi-bituminous	Pennsylvania	Cambria	Car run of mine	3.51	16.82	73.04	6.63	0.94	80.70	4.17	2.79	1.26	14,279	14,800
Semi-bituminous	West Virginia	Fayette	Car run of mine	2.52	21.28	68.20	8.00	0.68	78.71	4.45	4.33	1.31	13,997	14,359
Semi-bituminous	West Virginia	McDowell	Composite mine	3.50	15 0	78.30	3.20	0.60	84.39	4.20	2.73	1.38	14,800	15,330
Bituminous	Alabama	Bibb	Mine	3.16	31.05	59.56	6.23	1.20	78.28	4.98	4.78	1.37	14,141	14,602
Bituminous	Colorado	Boulder	Car run of mine	18.68	34.88	40.45	5.99	0.55	57.46	3.99	12.18	1.15	10,143	12,472
Bituminous	Illinois	Franklin	Mine	11.50	26.70	52.67	9.13	0.60	65.40	4.01	8.28	1.08	11,416	12,893
Bituminous	Illinois	La Salle	6 in. lump car	12.39	36.89	41.80	8.92	3.92	61.29	4.47	8.01	1.00	11,399	13,012
Bituminous	Illinois	Sangamon	Car run of mine	16.60	32.41	37.82	13.77	4.05	53.89	3.77	7.63	0.91	9,940	11,833
Bituminous	Illinois	Williamson	Mine	7.88	31.20	49.89	11.03	2.99	66.50	4.26	6.30	1.04	11,741	12,746
Bituminous	Indiana	Green	Car run of mine	10.30	36.31	41.64	11.75	4.23	61.00	4.24	7.42	1.06	11,218	12,508
Bituminous	Iowa	Marion	Car run of mine	14.21	33.17	37.40	15.22	4.66	54.08	3.93	6.59	1.31	10,019	11,678
Bituminous	Kansas	Crawford	Car run of mine	4.99	32.68	49.36	12.97	4.28	67.34	4.43	4.91	1.08	12,242	12,884
Bituminous	Kentucky	Hopkins	Car run of mine	7.92	36.09	45.93	10.06	3.52	65.29	4.51	7.30	1.40	12,022	13,055
Bituminous	Ohio	Hocking	Mine	9.72	32.44	53.41	4.43	0.54	69.50	4.62	9.94	1.25	12,247	13,567
Bituminous	Oklahoma	Pittsburg	Mine	2.09	27.59	50.25	20.07	5.73	63.66	4.23	2.89	1.33	11,695	11,945
Bituminous	Pennsylvania	Fayette	Car run of mine	5.13	27.87	58.29	8.71	0.86	73.13	4.34	6.33	1.50	13,365	14,089
Bituminous	Pennsylvania	Westmoreland	Car run of mine	3.39	31.79	56.46	8.36	1.05	74.42	4.69	6.70	1.39	13,699	14,178
Bituminous	Tennessee	Campbell	Car run of mine	5.38	34.54	53.03	7.05	0.99	72.41	4.90	7.55	1.72	13,048	13,788
Bituminous	Virginia	Lee	Car run of mine	4.06	34.93	56.28	4.73	1.20	76.59	4.87	7.31	1.24	13,826	14,411
Bituminous	West Virginia	Kanawha	Car run of mine	2.82	32.20	56.95	8.03	1.38	76.74	4.85	4.84	1.34	13,766	14,166
Bituminous	West Virginia	Mingo	Car run of mine	2.86	33.23	58.08	5.83	0.67	78.38	4.69	6.14	1.43	14,105	14,521
Sub-bituminous	Colorado	Boulder	Mine	19.15	30.82	44.27	5.76	0.25	56.38	3.80	13.58	1.08	9,616	11,893
Sub-bituminous	Washington	King	Mine	17.51	31.17	38.55	12.77	0.37	52.11	3.90	12.26	1.08	9,331	11,312
Lignite	North Dakota	McLean	Lump	35.96	31.92	24.37	7.75	1.15	41.43	2.54	9.96	1.21	7,069	11,038
Lignite	North Dakota	Williams	Car run of mine	38.92	25.54	30.15	5.39	0.48	39.34	2.56	12.63	0.68	6,739	11,034
Lignite	Texas	Wood	Car run of mine	33.71	29.25	29.76	7.28	0.53	42.52	3.04	12.13	0.79	7,348	11,084
Lignite	Wyoming	Sheridan	Mine	29.33	28.58	36.42	5.67	1.21	45.09	3.01	14.80	0.89	7,783	11,012

The suitability of a coal for storage purposes is influenced largely by the impurities contained within the coal, and also by the size of the coal. Safety in storing coal may be achieved by the proper selection of the coal, by storage under water, or by proper piling. When stored in piles the coal should be tightly compacted and the fines mixed in with coarser material; there should not be regions of coarse material adjacent to regions of fine material; air pockets and air circulation within the pile, as well as all sources of heat, should be avoided. The piles of coal should not be made too deep, and the temperatures of their interiors should be watched closely, as excessive or dangerous temperatures will necessitate re-piling. Coals high in oxygen and sulphur are harder to store satisfactorily.

59. Pulverized Fuels.—Coal, coke, and lignites when finely crushed and ground to a powder of the fineness of dust will float in the air and may be burned somewhat like a gaseous fuel. The limits of fineness are that all the powdered fuel will pass through a 50-mesh screen, and 75 to 80 per cent through a 200-mesh screen. The foregoing values vary somewhat with the fuel used. In general, the finer coal is ground the better will be its combustion. However, the extra cost of very fine pulverization may exceed the value of the benefits secured, so that there is an economical fineness as a limiting factor.

60. Wood. –Wood as a fuel is not of general commercial importance, and its use is confined to more or less isolated conditions. The moisture content of wood is variable, being as high as 50 per cent when freshly cut and from 10 to 20 per cent after having been exposed to the air for some time. Woods are classified as *hard* and *soft*, the heat values ranging from 8300 to 9000 B.t.u. per lb. of combustible. The desirability of a wood, as a fuel, is influenced by its moisture content and its density or weight per cubic foot. As woods are usually sold by volume instead of weight, the heat value per unit of measure varies with the dryness and the density of the air-dried wood. On a dry basis, nearly all woods show similar ultimate analyses which have the following ranges: carbon, 48 to 52; hydrogen, 5.9 to 6.5; oxygen, 41 to 45; nitrogen, 0.04 to 0.10; and ash, 0.20 to 1.0 per cent.

61. Refuse and Residue Solid Fuels.—In the lumber and woodworking industries, refuse woods used for steam generation include sawdust, slabs, shavings, and "hogged wood." The latter is wood refuse that has been cut to somewhere near uniform size by a special machine. This processing permits convenience and efficiency in handling and burning the refuse. The statements of Art. 60 apply to refuse wood fuels in regard to moisture content and heating values.

Tanbark is the remains of the oak or hemlock bark used in leather tanning. This material is in finely divided form and usually requires

the use of some other fuel, such as coal, to keep it burning. The moisture content may exceed 60 per cent, and the heating value may not run over 2500 to 3000 B.t.u. per lb. as fired.

Bagasse is the crushed stalks of sugar-cane from which juice has been extracted. The ultimate analyses of dry bagasse fall within the lower ranges of those of wood, and the heating values on a dry basis are within the limits given for wood. As the material usually contains a high percentage of moisture the heating values as received may be from 3500 to 4500 B.t.u. per lb.

Coke is the solid residue left after the destructive distillation of either petroleum oils or certain kinds of bituminous coals. In the former case the term petroleum coke is applied to the carbon deposit left in oil stills. The physical and chemical properties of either coal coke or petroleum coke are dependent upon (1) the specific kind of coal or oil used, (2) the method by which the volatile materials are driven off, and (3) the subsequent exposure to moisture.

Not all coals are suitable for the production of coke. Coal coke is produced as a necessary fuel for metallurgical processes and also as a by-product in the manufacture of artificial gas. Coal cokes consist largely of the carbon and the ash of the coal from which they are manufactured, together with small amounts of volatile material. The moisture content is generally greater than that of petroleum cokes.

Petroleum cokes have high carbon content and small amounts of ash. The percentages of volatile matter are low and are somewhat dependent upon the process of manufacture. The heating values are high, being somewhere in the neighborhood of 15,000 B.t.u. per lb. as received.

The analyses of coal and petroleum cokes vary; therefore, no single analysis may be taken as applicable to all cases. The analyses given in Table X are for samples of coal coke and petroleum coke suitable for use as fuels for steam power plants.

TABLE X

Analyses of Coke Breeze

Proximate Analysis as Received			Ultimate Analysis as Received		
Sample	Coal	Petroleum	Sample	Coal	Petroleum
Moisture	10.77	5.50	Carbon	73.16	87.01
Volatile matter	4.92	5.82	Hydrogen	1.18	3.91
Fixed carbon	72.26	87.50	Oxygen	1.13	0.63
Ash	12.05	1.18	Nitrogen	0.92	0.85
Heating value as received, B.t.u. per lb.	11,160	15,000	Sulphur	0.79	0.92
Heating value dry, B.t.u. per lb.	12,506	15,886			

In general the larger sizes of coke are used for either special purposes or for domestic heating, and only the smaller sizes are utilized for the production of steam in power plants. Coke screenings containing pieces $\frac{1}{2}$ in. and smaller in size are known as coke breeze. Such material may be burned upon a chain-grate stoker provided ignition can be maintained and an excessive amount does not fall through the grate. A very satisfactory method of burning the fine sizes of coke is in a pulverized form.

62. Fuel Oils.—Fuel oils are derived from petroleum oils which are found in many parts of the world in varying quantities. Outside of the United States the greatest production of crude petroleum occurs in Russia and Mexico. Within the United States the principal fields are: the Pennsylvania-Appalachian; the Lima, Indiana, Illinois; the Mid-continent; the Gulf; and the Western.

Crude petroleums are of uncertain origin but are of an organic nature. Such natural oils vary according to the base or residue left after distillation and are classified as: (1) paraffin base; (2) asphalt base; and (3) mixed base containing both paraffin and asphalt. In the United States the Eastern oils are generally paraffin base; the Mid-continent oils mixed base; and those of the Gulf and Western fields asphalt base. Those with asphalt bases provide most of the fuel oils.

The chief constituents are carbon and hydrogen, which are united in a series of very complex hydrocarbon compounds. Other items in the analyses are oxygen, nitrogen, sulphur, and moisture. The latter is very undesirable, especially when it forms an emulsion. Typical properties of a few crude and fuel oils are given by Table XI. In general the

TABLE XI

ANALYSES OF TYPICAL AMERICAN OILS *

Location	Physical Properties				Chemical Properties				
	Gravity A.P.I. at 60 Deg. F.	Specific Gravity at 60 Deg. F.	Flash Point, Deg. F.	Burning Point, Deg. F.	C Carbon	H Hydrogen	O Oxygen	S Sulphur	Calorific Value, B.t.u. per Lb.
Calif. crude midway	16.3	0.957	172	210	86.58	11.61	0.74	0.82	18,613
Pennsylvania crude	39.8	0.826	.....	.....	82.00	14.80	3.20†	.....	17,930
West Va. crude	36.7	0.841	.....	.....	84.36	14.10	1.60†	.....	18,400
Texas crude	22.3	0.920	142	181	84.60	10.90	2.87†	1.63	19,060
Oklahoma crude	25.3	0.903	264	286	87.93	11.37	0.19	0.41	19,650
Texas fuel	21.3	0.926	216	240	83.26	12.41	3.83	0.50	19,481
Pennsylvania fuel	28.2	0.886	.....	.....	84.90	13.70	1.40†	.....	19.200

* "Fuel Oil in Industry," by Stephen O. Andros.

† Includes nitrogen.

percentages of the principal components range as follows: carbon, 80 to 87; hydrogen, 12 to 15; and sulphur, 0 to 5. The specific gravities vary from 0.80 to 0.97, and the heating values from 18,500 to 20,000 B.t.u. per lb.

Some crude petroleum oils are worthless for the production of gasoline and are used without treatment as fuels. Most fuel oils are distillates, residues, or blends left from the distillation or cracking of petroleum in the manufacture of gasoline. The chemical composition and the calorific values of fuel oils are much the same as those of the original crudes, but these vary with the oils used and the blends made of various distillates. Fuel oils from which the more volatile constituents have been removed are much safer to handle than unrefined petroleums.

Fuel oils are standardized in six grades which are designated by numbers. Grades No. 1, 2, and 3 are known as light, medium, and heavy domestic fuel oils. Oils classified by the numbers 4, 5, and 6 are respectively light, medium, and heavy industrial fuel oils. Oils No. 5 and 6 correspond to the Federal specifications for Bunker Oil "B," and Bunker Oil "C."

In comparison with coal, oil as a fuel has the following advantages: (1) cleanliness, (2) flexibility of operation, (3) elimination of ash handling, (4) reduction of labor in firing, (5) ease of control, (6) high operating efficiency, (7) reduced maintenance charges on boilers, but not on furnaces, and (8) reduced standby losses.

The following disadvantages are to be considered in the use of oil: (1) initial cost of the storage tanks, burners, heaters, and pumps; furnace upkeep greater, (2) inadequacy of the supply, and (3) the cost of the supply.

63. Fuel Oil Specifications. In any event, whether the fuel oil be a crude, a topped crude, a distillate, or a blend, it should meet certain specifications. Fuel oils should be hydrocarbon oils free from grit, acid, and fibrous materials, and they should have limitations on their sulphur, silt, and moisture contents. Specifications for fuel oils should include the items of viscosity, specific gravity, flash point, burning point, pour point, calorific value, and water, silt, and sulphur contents.

Viscosity is a measure of the internal friction of the oil or its resistance to free flowing. Viscosity is specified at definite temperatures as determined by some specific viscosimeter. The viscosity determination is the time in seconds required for a definite quantity of the oil, at the prescribed temperature, to flow through the orifice of the viscosimeter.

Specific gravity is the ratio of the weight of a given volume of the oil to the weight of an equal volume of water at the same temperature. A temperature of 60 deg. F. is commonly used, so that sp. gr. $60^\circ/60^\circ$ F.

means that oil at 60 deg. F. has been referred to water at 60 deg. F. Specific gravities may be determined by use of either a hydrometer or a Westphal balance. The oil industry uses the American Petroleum Institute (A.P.I.) scale to express specific gravity in degrees. The A.P.I. scale fixes a reading of 10 deg. as equal to a specific gravity of 1.00. Readings greater than 10 deg. indicate a specific gravity less than 1.0 or an oil which is lighter than water. Actual specific gravities may be obtained from degrees A.P.I. by the following equation:

$$\text{Sp. gr.} = \frac{141.5}{131.5 + \text{deg. A.P.I.}}. \tag{62}$$

Specific gravities are useful in computing the weights of unit volumes, as oils are usually sold either by the volume in gallons or barrels of 42 gallons.

Flash point and *burning point* are, respectively, the temperatures at which, when an oil is heated, its vapors flash or burn momentarily, and burn continuously when a small flame is brought near its surface. *Pour point* is the lowest temperature at which cold oil will flow.

Calorific determinations are made as indicated in Art. 67. Actual determinations of heating value are the most useful, as the calorific value may be only approximately calculated by Dulong's equation.

Water and sediment contents should not exceed 2 per cent by volume. Although sulphur is undesirable, an amount equal to 4 per cent is not objectionable.

64. Gaseous Fuels.—The use of gas as a fuel in internal-combustion engines is very common. In the production of steam the application is more limited and the use is generally confined to natural, blast-furnace, and coke-oven gases, analyses of which are given by Table XII. One outstanding example of a plant using natural gas as a steam fuel is the Deepwater Plant at Houston, Texas, which develops a steam pressure of 1400 lb. per sq. in.

Natural gas occurs in various localities and usually consists of a high percentage of methane (a carbon-hydrogen compound), ethane, and carbon monoxide, and small percentages of nitrogen, carbon dioxide, etc. The heat value of natural gas ranges from 850 to 1150 B.t.u. per cu. ft.

Blast-furnace gases are low in heating values, which range from 80 to 120 B.t.u. per cu. ft. The heat values are due largely to the carbon monoxide (CO) content, which is about 25 to 35 per cent. Other constituents are hydrogen, carbon dioxide (CO_2), nitrogen, and a very small percentage of oxygen.

By-product coke-oven gases range from 400 to 600 B.t.u. per cu. ft. in heat value and are much more valuable than blast-furnace gases.

They contain considerable percentages of hydrogen and methane, together with low percentages of carbon dioxide, nitrogen, oxygen, and carbon monoxide.

TABLE XII

ANALYSES OF FUEL GASES

Gas	Composition by Volume, Per Cent								Heating Value B.t.u. per Cu. Ft.
	Hydrogen, H_2	Methane, CH_4	Ethylene, C_2H_4	Ethane, C_2H_6	Carbon monoxide, CO	Carbon dioxide, CO_2	Oxygen, O_2	Nitrogen, N_2	
Natural, Penna.	.....	90.0	.....	9.0	.....	0.2	.....	0.8	1126
Okla.	.....	94.3	.....	.....	.....	1.10	.....	4.6	1006
Cal.	.....	59.2	.....	13.9	.....	26.2	.....	0.7	889
Blast-furnace	1.0	.....	.....	.....	27.5	11.0	0.5	60.0	91
	3.2	.....	.....	.....	26.2	13.0	.....	57.6	93
Coke-oven	50.0	36.0	4.0	.....	6.0	1.5	0.5	2.0	603
	57.4	28.5	2.9	.....	5.1	1.4	0.5	4.2	536

65. Calorific Value of Fuels.—The heat liberated by the complete and rapid oxidation (or burning) of a unit weight or volume of a fuel is the *heating* or *calorific value* of the fuel. For solid and liquid fuels the heating value is expressed in B.t.u. per lb. For gaseous fuels the heating values are expressed in B.t.u. per cu. ft. of gas measured at a temperature of 60 deg. F. and a pressure of 30 in. of mercury.

Fuels which contain hydrogen or hydrocarbons have two heating values, the *higher* heating value, and the *lower* heating value. The combustion of hydrogen produces superheated water vapor which in the ordinary furnace escapes at the temperature of the chimney gases. Consequently, the superheat and the heat of evaporation of the vapor formed are not available for heating purposes. The *lower heating value* is the net heat liberated per pound of fuel after the heat necessary to vaporize and superheat the steam formed from the hydrogen, as well as that from the fuel, has been deducted. The *higher heating value* of a fuel is the one indicated by any sort of a fuel calorimeter. In the case of calorific determinations, the steam produced in the calorimeter is condensed and the heat of superheat and the heat of evaporation are recovered.

The heating value of a fuel may be found by two methods: by calculation, and by the use of a fuel calorimeter. When the ultimate analysis of the fuel is known the heating value may be computed. The most satisfactory method of determining the heating value is to burn a quan-

tity of the fuel in a calorimeter and to measure the rise in temperature induced in a given quantity of water. When the proper corrections are applied for the calorimeter parts, radiation, etc., the heat value per unit of the fuel may be determined from the calorimeter data.

66. Heating Value of a Fuel by Calculation.—In the calculation of the heating value of a coal from the ultimate analysis, only carbon, hydrogen, and sulphur are considered as heat producers. Not all the hydrogen indicated by the ultimate analysis is available for the production of heat. A part of the hydrogen of the fuel is rendered unavailable for heating by the oxygen which is in combination with it in the form of inherent moisture. The hydrogen available for the production of heat is $\left(H - \frac{O}{8}\right)$, see Art. 76. H and O represent the proportionate parts of hydrogen and oxygen, by weight, per pound of the fuel. The heats liberated by the complete combustion of one pound of each of the following elements are: carbon, 14,600 B.t.u.; hydrogen, 62,000 B.t.u.; and sulphur, 4050 B.t.u.

Dulong's formula for the heating value of a fuel based on the proportions by weight of the total carbon, the available hydrogen, and the sulphur is expressed, in B.t.u. per pound, by

$$F = 14{,}600C + 62{,}000\left(H - \frac{O}{8}\right) + 4050S. \qquad (63)$$

Only for coals of some regions are the heat values calculated by the use of Dulong's equation in close agreement with those obtained by the use of a calorimeter. Certain coals, such as Illinois coals, may contain calcium carbonate, which may lead to an erroneous ash determination so that the percentages of oxygen and hydrogen are also incorrect. A discrepancy between the calculated and the calorimeter-determined heating value results.

Dulong's equation gives only approximate results when used to calculate the heating value of a liquid fuel from the ultimate analysis. The true calorific value of a liquid fuel is most accurately determined by use of a fuel calorimeter.

Gaseous fuels containing hydrogen or hydrocarbons have a high heat and a low heat value, depending upon whether or not the heat of the vapors in the products formed is recovered. Ordinarily in engineering work the high heat values are used.

The calculation of the heating value of gaseous fuels may be made from their volumetric analyses if the calorific values per cubic foot of the various constituents are known. Such calculations are based on the assumption that each constituent is free and does not influence the

others. The calorific values of several constituents of gases are given in Table XIII.

TABLE XIII

CALORIFIC VALUES OF GASES

At 60 deg. F. and 29.92 in. of mercury

Gas	Symbol	Heating Value per Cubic Foot	
		Higher	Lower
Carbon monoxide	CO	322	322
Hydrogen	H_2	330	282
Methane	CH_4	1008	916
Ethylene	C_2H_4	1598	1510
Ethane	C_2H_6	1763	1625

Example.—Calculate the higher heating value, F, of one cubic foot of natural gas having the following volumetric analysis: C_2H_6, 9.0; CH_4, 90.0; CO_2, 0.2; and N_2, 0.8 per cent.

Solution.—The ethane and the methane are the heat-producing compounds and have a higher heat value of 1763 and 1008 B.t.u. per cu. ft., respectively.

$$F = (1763 \times 0.09) + (1008 \times 0.9) = 1066 \text{ B.t.u. per cu. ft.}$$

67. Solid- and Liquid-fuel Calorimeters.—Fuel calorimeters are of two general types: those which employ the oxygen bomb, and those which utilize sodium peroxide as an agent to supply the oxygen for combustion within the closed cartridge of the calorimeter. The latter type of calorimeter gives results that are satisfactory for many purposes, but the oxygen-bomb type is recognized as the standard calorimeter in the determination of the heating value of fuels. Standard calorimeters are the Mahler, Atwater, Emerson, and the Parr oxygen bomb calorimeter. It is essential that any oxygen bomb calorimeter should have a non-corrosive bomb or a bomb with a perfect non-corrosive lining.

The cross-section of a Parr oxygen bomb is shown by Fig. 23. The essential parts of the bomb are the cover A, the cover clamp nut C, the supports DD, the fuel tray G, the rubber gasket B, and the check valve E. The supports DD serve as electrodes, and a fine fuse wire is looped between them as shown so that it dips into the fuel held in the fuel tray G. The bomb is capable of holding oxygen under a presssre of 350 to 450 lb. per sq. in.

The assembled bomb, containing the fuel and oxygen under 20 to 30 atmospheres pressure, is placed in the calorimeter as shown in Fig. 24. Part *H* is a metal water container of oval cross-section. The bomb *F* is placed slightly to one side of the center of the oval can so that the motor-driven water stirrer *S* may have space. The thermometer *T*, for the measurement of the temperature rise of the water surrounding the bomb, is placed at the opposite side of the oval can. The body of the calorimeter, including the cover and bottom, is made of indurated fiber with an air space in the walls. This body provides considerable protection against a flow of heat to the calorimeter water from the sur-

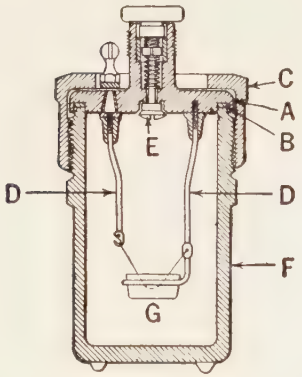


Fig. 23.—Oxygen bomb details.

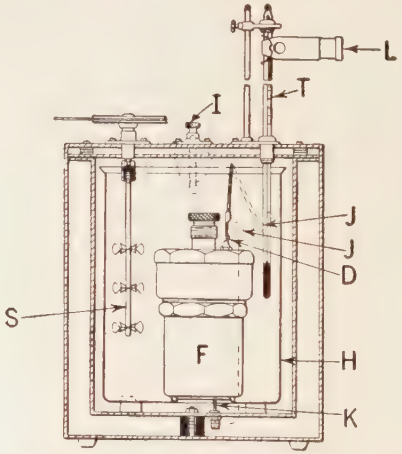


Fig. 24.—Oxygen bomb calorimeter assembly.

rounding air or a loss of heat from the water to the surrounding air. Part *I* represents one of the two clamps used for holding the cover in place. The electrical circuit through the fuse wire is completed through the binding posts *JJ*. An external source of low-voltage current is applied. The shell of the bomb *F* makes electrical contact at *K*, and a flexible electrical connection is attached to one electrode *D* at *J*.

A determination of heating value is made with the calorimeter in the following manner. A carefully weighed amount, approximately one gram, of finely pulverized air-dried coal is placed in the fuel tray *G*. A loop of No. 34 B.S. gage pure-iron wire is attached to the electrodes so that this fuse wire just touches the fuel. After the bomb cover is in place the bomb is carefully and slowly charged with oxygen.

The bomb is then placed in the oval container located within the calorimeter body, and 2000 grams of distilled water are added to the can. The water-stirring mechanism is set in motion, and preliminary temperature observations are taken with the calorimeter thermometer. The fuel

is ignited by closing the electrical circuit for a very short period of time. Observations of the water temperature are recorded until a period of five minutes after the end of the combustion period has elapsed. The preliminary and final temperature observations are used to determine the radiation corrections.

The total temperature rise of the calorimeter water, corrected for radiation effects, is multiplied by the weight of water including the water equivalent of the metal parts of the calorimeter bomb and can. The water weight and the water equivalent of the calorimeter are best expressed in pounds. If the thermometer readings are in degrees Fahrenheit, the product of the equivalent water weight and the temperature rise represents the total heat in B.t.u. liberated within the calorimeter. Additional corrections are necessary to compensate for the heat produced by the formation of nitric and sulphuric acids, and also the combustion of the fuse wire. From these derived data the heat value per pound of the fuel may be computed.

The procedure in the case of calorific determinations of liquid fuels is the same as for solid fuels. However, in the case of volatile liquid fuels it is necessary to weigh the fuel in closed glass bulbs, which are broken in the calorimeter fuel tray just before the bomb is closed. The loss of the volatile constituents of the fuel is thus prevented during the weighing operation.

68. Gas Calorimetry.—A typical gas calorimeter is the Sargent calorimeter, shown by Fig. 25. The gas whose heat value is to be determined is passed through a calibrated meter D and its pressure and temperature noted. A pressure regulator in the meter outlet line maintains the pressure of the gas nearly uniform as it flows to a carefully adjusted Bunsen burner B located within the calorimeter body. The calorimeter body C is a vertical cylindrical chamber having a water jacket through which water is allowed to flow under a constant head. Calibrated thermometers TT are placed in the inlet and outlet water connections. The heat produced by the gas is determined by accurately weighing the water passing through the calorimeter while a given quantity of gas is burned. Frequent readings of the inlet and outlet water temperatures are taken during this period and the average temperature rise of the water is determined. The product of the temperature rise and the weight of water passing through the calorimeter, divided by the cubic feet of gas burned, gives the heating value of the fuel per cubic foot as measured. This result may be corrected to heating value per cubic foot at the standard conditions of 60 deg. F. temperature and 30 in. of mercury pressure. The heating value thus determined is the higher value. If the lower heating value is desired, it is necessary to collect the moisture condensed from the products of combustion and

make a correction for the heat liberated by the condensation of the vapor formed during combustion.

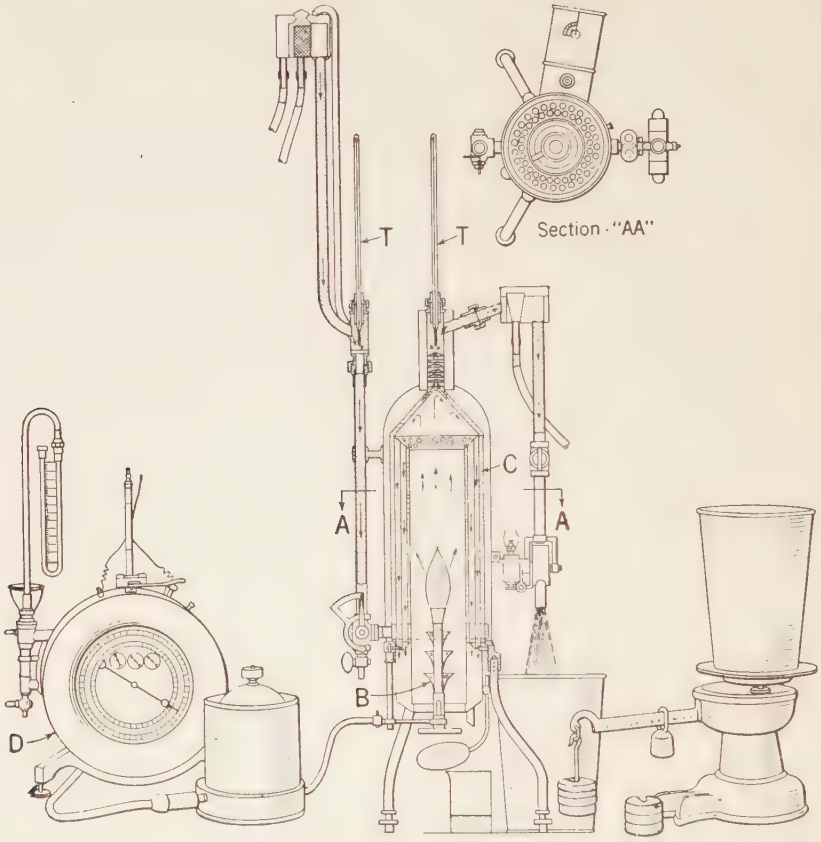


FIG. 25.—Sargent gas calorimeter.

PROBLEMS

1. Express the proximate analysis of a Sangamon County, Illinois, bituminous coal on a dry basis, and also on a moisture- and ash-free basis.

2. Calculate the proximate analysis of a Campbell County, Tennessee, bituminous coal on a dry basis and also on a moisture- and ash-free basis.

3. Express the proximate analysis of a McDowell County, West Virginia, semi-bituminous coal on a dry basis and on a basis of the combustible.

4. Determine the ultimate analysis, on a dry basis, for a Hocking County, Ohio, bituminous coal.

5. Find the ultimate analysis, on a dry basis, of a Schuylkill County, Pennsylvania, anthracite coal.

6. Determine the ultimate analysis, on a moisture- and ash-free basis, of a Wood County, Texas, lignite.

7. A bituminous coal contained 10 per cent of moisture. The proximate analysis on the dry basis was: volatile matter, 40.45; fixed carbon, 46.37; and ash, 13.18 per cent. Express the as-received analysis.

8. The heating value of the coal of problem 7 on a dry basis is 12,883 B.t.u. per lb. Find the heating value per pound of coal as received and also per pound of combustible.

9. Calculate by Dulong's equation the heating value per pound of Boulder County, Colorado, sub-bituminous coal.

10. By use of Dulong's equation calculate the heating value, on a dry basis, of one pound of McDowell County, West Virginia, semi-bituminous coal.

11. Calculate the heat that may be liberated by the combustion of one pound of Hopkins County, Kentucky, coal if 5 per cent of the carbon is burned to CO instead of CO_2.

12. A fuel oil has an A.P.I. gravity at 60 deg. F. of 27.1 deg. Find the weight of the oil in pounds per gallon.

13. Find the total heating value of one barrel (42 gallons) of fuel oil whose A.P.I. gravity at 60 deg. F. is 16.6 deg., and whose calorific value is 18,800 B.t.u. per lb.

14. How many barrels of the Texas fuel oil of Table XI are equivalent in heating value to one ton of Pittsburg County, Oklahoma, coal? How many tons of the coal are equivalent to 100 barrels of the oil? Neglect thermal losses.

15. A No. 6 industrial fuel oil had a heating value of 150,000 B.t.u. per gallon and when it was burned 20 per cent of its heating value was lost in various ways. How many gallons of the oil were equivalent to 25 tons of coal whose heating value is 11,200 B.t.u. per lb. as fired, if 65 per cent of the heat of the coal can be utilized? If the heating value of the oil was 19,500 B.t.u. per lb., what was its A.P.I. gravity?

16. Calculate the higher heating value in B.t.u. per cu. ft. of a natural gas having the following analysis by volume: CH_4, 74.10; C_2H_6, 17.87; CO_2, 0.14; O_2, 0.0; and N_2, 7.89 per cent.

17. Find the higher heating value of a by-product coke-oven gas whose analysis by volume was: H_2, 56.9; CH_4, 22.6; C_2H_4, 3.0; CO, 8.7; CO_2, 3.0; and N_2, 5.8 per cent.

18. A pipe line 500 miles long having an internal diameter of 24 inches is used to convey natural gas at an average absolute pressure of 300 lb. per sq. in. and a temperature of 50 deg. F. The heating value of the gas at standard conditions is 1000 B.t.u. per cu. ft. What is the heating value of the gas held in the pipe?

19. A sample of coal lost 6.2 per cent of its weight when air-dried. A quantity of the air-dried coal weighing 0.95 gram was burned in an oxygen-bomb calorimeter whose water equivalent was 2430 grams. The total temperature rise of the water in the calorimeter was 5.41 deg. F., of which a temperature rise of 0.31 deg. F. was produced by the fuse wire, acid formation, etc. Find the heating value of the coal in B.t.u. per lb. as received.

20. City gas measured at a temperature of 77 deg. F. and a gage pressure of 5.2 inches of water was burned in a gas calorimeter to give the following test data: temperature of inlet water, 58.2 deg. F.; temperature of outlet water, 75.6 deg. F.; weight of water used, 6.26 lb.; gas used, 0.20 cu. ft.; and barometer, 29.30 inches of mercury. Find the heating value of the gas per cu. ft. at standard conditions.

CHAPTER VI

COMBUSTION AND FLUE-GAS ANALYSIS

69. The Combustion Process.—The three principal heat-producing elements of fuels are carbon, hydrogen, and sulphur, separately or in combination. The reaction that releases heat energy from the elements is one of oxidation, in which the necessary oxygen is supplied from the air. Unless the process is a rapid one, the amount of heat energy released in a given period of time is small. Therefore, in the combustion of fuels for power purposes the oxidation processes must be rapid and must produce a considerable quantity of heat. Where the fuel exists in a dust, gaseous, or atomized form, under certain conditions, exceedingly rapid oxidation or combustion may produce an explosion. In general, the combustion reactions take place when the burnable parts of the fuel have been reduced to a gas and unite with the oxygen of the air for further oxidation. Except in specific cases this process is slow enough to prevent an explosive action.

70. Combustion Prerequisites.—Three conditions are necessary to produce and maintain the complete combustion process of fuels. The first requirement is an adequate supply of oxygen procured from a sufficient but not excessive air supply. An inadequate quantity of air provides a deficient amount of oxygen; an excessive amount of air produces a large volume of flue gases with a resultant loss of heat to the chimney. Second, the mixture of the oxygen and the burnable parts of the fuel must be thorough. Third, the temperature maintained must be high enough to gasify the fuel, to ignite the mixtures, and to maintain the process.

Atomization or vaporization of liquid fuels is the first essential in their combustion, together with sufficient air and the proper ignition temperature. Unless the atomization is thorough the combustion will be incomplete and an inefficient process results.

71. Gaseous Products of Combustion. –Pure carbon when burned with an adequate and thorough mixture of air will liberate 14,600 B.t.u. per pound of carbon. The resulting gas formed is carbon dioxide, CO_2. If the air supply is deficient or if a thorough mixture of the carbon and the air is not obtained, the gas formed will be carbon monoxide, CO. When the latter condition exists, only 4440 B.t.u. are liberated per

pound of carbon and the process is an inefficient one in furnace operation, owing to the loss of a valuable unburned portion of the fuel.

Pure hydrogen burns to water vapor. Hydrogen alone or in combination with carbon (hydrocarbons), may escape if the combustion process is incomplete. Hydrocarbons completely burned disappear as water vapor and carbon dioxide. Sulphur usually burns to sulphur dioxide, SO_2, in the ordinary furnace.

Moisture in the coal is converted into water vapor or superheated steam and usually represents a heat loss. In the case of fuels of very fine sizes, and where the draft is heavy, moisture may be an aid in securing higher operating efficiencies, as unburned fuel particles may be prevented from being discharged at the stack. Moisture may also cause certain coals to cake, thus reducing the amount of fine unburned coal appearing in the ash and refuse, and producing a resultant saving.

72. Combustion Stages.—Combustion proceeds as follows when a solid fuel such as coal is burned upon a grate. Immediately after a fresh charge is thrown upon the incandescent fuel bed the moisture disappears as steam. The volatile matter, which includes hydrogen and hydrocarbons, next distills off, and if its combustion is complete, water vapor and carbon dioxide are formed. These products escape, together with sulphur dioxide, and the oxygen and nitrogen are released from the fuel. If the combustion of the volatile matter is incomplete, unburned hydrocarbons may escape as well as water vapor, carbon monoxide, and carbon dioxide. The remnants of the fuel left, after the moisture and the volatile matter have disappeared, are the fixed carbon and the ash.

The fixed carbon in the lower part of the fuel bed, where the air supply is sufficient, immediately burns to carbon dioxide. However, as this CO_2 passes upward through the fuel bed, the oxygen supply is generally deficient in proximity to the large quantities of carbon. As a consequence, a reducing instead of an oxidizing action takes place and the carbon dioxide is changed to carbon monoxide. This carbon monoxide, as it passes from the fuel bed, is again oxidized to carbon dioxide, provided the air supply is sufficient and a thorough air-gas mixture is secured at a temperature sufficient for ignition. Sometimes it is necessary to admit air above the fire in order to secure efficient combustion.

An inadequate supply of air, a small furnace volume, and cold surfaces or low temperatures above the fire hinder further combustion and promote the formation of soot, smoke, and carbon monoxide, as well as the escape of unburned hydrogen and hydrocarbons.

73. Furnace Requirements.—The design of the furnace plays an important part in the completeness of combustion secured. The

various fuels require different types of furnace construction and different boiler settings. Unless the furnace is properly designed, a thorough mixture of air and the gases from the fuel, and a sufficiently high temperature for their combustion, cannot be maintained.

Adequate combustion space should be provided in the furnace, or in a chamber adjacent to it, to allow an intimate mixture of the gaseous products with the requisite amount of air. In modern large boiler installations the boilers are set high above the level at which the fuel is admitted to the furnace, and a large combustion space is provided within the furnace itself. In older boiler settings a refractory-lined combustion chamber is provided behind the bridge wall where the combustion process is completed. In such an arrangement the air and gases are diffused as they pass over the bridge wall. In the latter setting both the furnace and the combustion chamber should be separated from the boiler heating surfaces by refractory arches. Ordinarily the reactions of combustion must be completed before the gases leave the combustion chamber and strike the comparatively cool heating surfaces of the boiler. Secondary combustion, in and around the tubes of a boiler, is unsatisfactory and likely to be variable in its completeness.

An exception to the rule of not having comparatively cool surfaces for the unburned gases to play upon is found in the large modern stoker-fired or powdered-coal-fired boiler furnaces. These furnaces invariably are large in volume, and the boilers are driven at several times their normal rating. In such furnaces very high combustion temperatures are secured, so that the refractory linings of the furnaces are likely to be melted, or badly eroded by molten slag and ash from the coal. In such cases water pipes may be embedded in the furnace walls just behind the inner face. Water from the boiler is circulated through these pipes which serve as additional boiler heating surface. These water pipes are for the purpose of keeping the furnace linings below their melting point. Another method used in large modern boiler furnaces is that of the water wall which forms the side enclosure of the furnace. Water from the boiler is circulated through this additional heating surface. These water tubes have given good results even though they bring a comparatively cool surface in proximity to the burning fuel. The radiant type of steam superheater is also used for the same purpose. However, these devices are adapted only to large furnaces using preheated combustion air; moreover, the whole combustion process must be carefully handled.

Another prerequisite of a good furnace is the control of the air admitted to the furnace. A deficiency of air, or a great excess of air, supplied to the furnace results in loss of efficiency. The furnace must be tight, so that air cannot leak into it at points other than where air is

required. Air leakage may cause the furnace temperature to be reduced below the ignition temperature of the gaseous products, or else it may cause an excessive amount of heat to be carried away by the greatly diluted flue gases. When the flue gases are diluted with excess air, their temperature is lowered and the heat transmitted to the water of the boiler is less.

From the standpoint of efficient and thorough combustion, as well as rapid transfer of heat from the flue gases through the boiler tubes and plates to the water within the boiler, high furnace temperatures are desirable. On the other hand, high furnace temperatures may result in rapid deterioration of the furnace linings, with the resultant maintenance charges, and the unavailability of the boiler for use during furnace lining repairs.

Stoker-fired boiler furnaces of medium or large size should have a volume of $\frac{1}{3}$ to $\frac{1}{2}$ cu. ft. for each pound of fuel burned per hour. A fair value for the volume of pulverized fuel furnaces is 1 cu. ft. of internal volume for each pound of powdered coal burned per hour.

74. Combustion Chemistry.—The atoms of each chemical element have a definite and fixed mass or weight. The hydrogen atom is the lightest, and its weight has been taken as unity or 1. The atomic weights of the elements usually involved in combustion reactions are given in Table XIV. The numerical figures given are not the exact atomic weights of the elements but are sufficiently accurate for all combustion calculations.

TABLE XIV

NAMES, SYMBOLS, AND ATOMIC WEIGHTS OF ELEMENTS INVOLVED IN COMBUSTION REACTIONS

Name	Symbol	Atomic Weight	Name	Symbol	Atomic Weight
Hydrogen	H	1	Oxygen	O	16
Carbon	C	12	Sulphur	S	32
Nitrogen	N	14			

So long as the atoms of an element remain uncombined with the atoms of other elements, their physical and chemical properties are unchanged. When the atoms of an element enter into chemical combination with other atoms, the physical and chemical properties of the molecule formed are different from those of the molecules of the elements entering into the combination. The properties of the molecule formed are distinctive for the particular combination. The combinations of atoms to form molecules are dependent upon the conditions

existent at the time of the combination. Hence, the molecules formed may be predetermined by the proper control of the conditions attendant to their formation from atoms.

Avogadro's Law, deduced experimentally, shows that, *when at the same temperature and pressure, equal volumes of all gases contain the same number of molecules.*

A convenient unit of measurement is the pound-mol or *mol*, which is a weight in pounds equal to the molecular weight of the element or compound. A mol of all elements, or compounds, contains the same number of molecules. One mol of all gases occupies the same volume under standard conditions of 32 deg. F. and 29.92 in. of mercury abs. pressure. The volume of one mol of gas is the product of its molecular weight and the volume per pound of the gas. Thus the volume of one mol of oxygen, O_2, at standard conditions is $11.21 \times (16 \times 2) = 358.7$ cu. ft.

When a reaction is written in the molecular form, the coefficients represent the relative number of cubic feet of the substance entering into the reaction as well as the weights. In a mol-volume V of a mixture of three gases whose fractions of the total volume are v_1, v_2, and v_3, there are v_1 mol volumes of gas *1*, v_2 mol-volumes of gas *2*, and v_3 mol-volumes of gas *3*, when all exist at the same temperature and pressure. The total weight of the mol is mV, when m is the molecular weight of the gas mixture. If the molecular weights of the gas constituents are represented by m_1, m_2, and m_3, the weights of the various gases of the mixture are m_1v_1, m_2v_2, and m_3v_3. The total weight of the gas mixture is $mV = m_1v_1 + m_2v_2 + m_3v_3$, and the proportional parts of the mixture w_1, w_2, and w_3, by weight, are,

$$w_1 = \frac{m_1v_1}{mV},\ w_2 = \frac{m_2v_2}{mV} \text{ and } w_3 = \frac{m_3v_3}{mV}.$$

75. Formation of Carbon Dioxide and Carbon Monoxide.—Carbon when oxidized may burn to either carbon dioxide, CO_2, or carbon monoxide, CO, depending upon the amount of oxygen entering into the reaction. In the case of the formation of CO the combustion of the carbon is incomplete. The formation of carbon dioxide is represented by the equation

$$C + O_2 = CO_2, \tag{64}$$

in which one combining weight of carbon unites with two combining weights of oxygen to form carbon dioxide. If the atomic weights are substituted in equation 64, the molecular weight of the carbon dioxide is found to be

$$12 + (2 \times 16) = 44.$$

If the numerical equation is divided by 12, the atomic weight of carbon, the relative proportions of the components are

$$1 + 2\tfrac{2}{3} = 3\tfrac{2}{3},$$

which signifies that $2\frac{2}{3}$ lb. of oxygen are required for the complete combustion of each pound of pure carbon.

Air by weight contains 23.15 per cent oxygen and 76.85 per cent nitrogen and other inert gases. As air is the usual source of oxygen for combustion processes, the weight of air required for the complete combustion of one pound of carbon is $(2\frac{2}{3} \div 0.2315) = 11.52$ lb.

When carbon does not take up the maximum amount of oxygen in combustion and CO is formed, the relation of the carbon to the oxygen is

$$2C + O_2 = 2CO. \tag{65}$$

In this case two atoms of carbon unite with two atoms of oxygen to form two molecules of carbon monoxide CO. The weight of 2CO is

$$(2 \times 12) + (2 \times 16) = 56$$

and the relative proportions of the constituents are

$$1 + 1\tfrac{1}{3} = 2\tfrac{1}{3}.$$

Thus in the formation of CO only $1\frac{1}{3}$ lb. of oxygen are required per pound of carbon, and the weight of air necessary is

$$(1\tfrac{1}{3} \div 0.2315) = 5.76 \text{ lb.}$$

Carbon monoxide may be burned to CO_2 under the favorable conditions of an additional oxygen supply and a temperature sufficient to facilitate the process. In this case two molecules of the monoxide unite with one molecule of oxygen to form carbon dioxide.

$$2CO + O_2 = 2CO_2. \tag{66}$$

The process requires $(2\frac{2}{3} - 1\frac{1}{3}) = 1\frac{1}{3}$ lb. of additional oxygen per pound of carbon.

76. Hydrogen and Sulphur Combustion.—The burning of hydrogen to water vapor proceeds according to the following equations:

$$\begin{gathered} 2H_2 + O_2 = 2H_2O \\ 4 + 32 = 36 \\ 1 + 8 = 9. \end{gathered} \tag{67}$$

Thus the complete burning of one pound of hydrogen requires 8 lb. of oxygen, and 9 lb. of water vapor are formed. The air required is $(8 \div 0.2315) = 34.56$ lb. per pound of hydrogen consumed.

Sulphur burns in air to sulphur dioxide, SO_2, and the process is expressed

$$\begin{aligned} S + O_2 &= SO_2 \\ 32 + 32 &= 64 \\ 1 + 1 &= 2. \end{aligned} \tag{68}$$

Each pound of sulphur burned requires one pound of oxygen, so that the air requirements are $(1 \div 0.2315) = 4.32$ lb. per pound of sulphur completely burned.

77. Combustion Data.—The combustion reactions for several of the important constituents of fuels are given by Table XV. Included also are the weights of oxygen and air required together with the weight of gaseous products formed when combustion takes place with the theoretical amount of air. The volume of the air required and the volume of the gaseous products may be computed, for any temperature or pressure, by use of the gas equation $PV = MRT$ or $V = MRT/P$.

78. Theoretical Amount of Air for Combustion.—The theoretical amount of air for combustion may be computed from the ultimate analysis of a fuel. The parts of a solid or a liquid fuel available for heat production are the total carbon, the available hydrogen, and the sulphur. When the fractional parts, per pound of fuel, of each of the heat-producing constituents are multiplied by the air necessary for the complete combustion of a pound of that constituent, the sum of the air requirements is the theoretical weight of air necessary to burn one pound of the fuel.

$$W_{ta} = 11.52C + 34.56\left(H - \frac{O}{8}\right) + 4.32S. \tag{69}$$

Equation 69 is applicable when the fuel constituent percentages are expressed by weight. In the case of gaseous fuels the measurements of fuels and the percentages of the fuel analyses are based on volumes; therefore, it is necessary to calculate on a volume basis the theoretical air required.

The volumes of oxygen required for the combustion of burnable gases may be obtained by setting up the combustion reactions, and volume relations not given by Table XV may be determined. From such data the theoretical volume of air required may be calculated.

Example.—The analysis of a natural gas showed the following percentages by volume: ethane, 9.0; methane, 90.0; carbon dioxide, 0.2; and nitrogen, 0.8. Find the theoretical volume of air required per cubic foot of gas if the gas and the air are at the same temperature and pressure.

Solution.—In this gas the C_2H_6 and the CH_4 are combustible. From Table XV it may be seen that each volume of C_2H_6 requires $7 \div 2$ or 3.5 volumes of

TABLE XV

COMBUSTION DATA

Combustible	Symbol	Reaction	Product	Volumes	Weights per Pound of Combustible				Heating Value, B.t.u. per Lb.
					Oxygen, Lb.	Nitrogen, Lb.	Air, Lb.	Gaseous Products, Lb.	
Carbon	C	$C + O_2 = CO_2$	Carbon dioxide	1 vol. C + 1 vol. O_2 = 1 vol. CO_2	2.667	8.85	11.52	12.52	14,600
Carbon	C	$2C + O_2 = 2CO$	Carbon monoxide	2 vol. C + 1 vol. O_2 = 2 vol. CO	1.333	4.43	5.76	6.76	4,440
Carbon monoxide	CO	$2CO + O_2 = 2CO_2$	Carbon dioxide	2 vol. CO + 1 vol. O_2 = 2 vol. CO_2	0.571	1.90	2.47	3.47	10,160
Hydrogen	H	$2H_2 + O_2 = 2H_2O$	Water	2 vol. H_2 + 1 vol. O_2 = 2 vol. H_2O	8	26.56	34.56	35.56	62,000
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	Carbon dioxide and water	1 vol. CH_4 + 2 vol. O_2 = 1 vol. CO_2 + 2 vol. H_2O	4	13.28	17.28	18.28	23,850
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	Carbon dioxide and water	1 vol. C_2H_4 + 3 vol. O_2 = 2 vol. CO_2 + 2 vol. H_2O	3.43	11.38	14.81	15.81	21,600
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	Carbon dioxide and water	2 vol. C_2H_6 + 7 vol. O_2 = 4 vol. CO_2 + 6 vol. H_2O	3.733	12.40	16.13	17.13	22,230
Sulphur	S	$S + O_2 = SO_2$	Sulphur dioxide	1 vol. S + 1 vol. O_2 = 1 vol. SO_2	1	3.32	4.32	5.32	4,050

oxygen and each volume of CH_4 requires 2 volumes of oxygen. Therefore, the volume of oxygen required per cubic foot of gas is $(0.09 \times 3.5) + (0.9 \times 2) = 2.115$ cu. ft. Dry air is composed of 20.99 per cent of oxygen and 79.01 per cent of nitrogen and other inert gases by volume. Hence, the volume of air required is $2.115 \div 0.2099 = 10.07$ cu. ft.

79. Composition of Flue Gases.—Gaseous combustion products, which pass from the furnace or boiler setting to the chimney, are known as flue gases. The fuel burned, the conditions of combustion, and the tightness of the furnace and boiler setting affect the composition of the flue gas.

Flue gases produced by the burning of a solid fuel, such as coal, usually contain percentages of carbon dioxide, carbon monoxide, oxygen, nitrogen, free carbon, ash, unburned hydrocarbons, sulphur dioxide, and superheated steam. If the combustion process has been absolutely complete, free carbon, carbon monoxide, hydrogen, and hydrocarbons will be absent.

Difficulty is encountered in controlling the combustion process so that it is complete with just the theoretical amount of air. Some excess air is required to insure complete combustion; thus oxygen usually appears in flue-gas analyses in varying amounts. Free unburned carbon and ash dust, appearing in a finely divided state in the flue gases, give the gases colors ranging from a light gray to deep black. The term smoke is then applied to the flue gases. The appearance of free carbon in flue gases is due to the fact that, even though ample air has been provided, the gases have been below the ignition temperature before complete oxidation of all the burnable parts has been completed. This is especially true of coals or fuels containing oily or tarry compounds. Separately, the free carbon particles of smoke do not represent a great heat loss, but frequently the completeness of the combustion process is indicated by the smoke color. Dark to black smoke is indicative of carbon monoxide and unburned hydrocarbons. The two latter items represent a greater heat loss in the chimney gases than soot or free carbon.

80. Flue-gas Analyses. Flue-gas analyses are based on the percentages of the various constituents by volume in the dry gases. Usually the analyses include only the percentages of carbon dioxide, oxygen, carbon monoxide, and nitrogen. More complete analyses include the percentages of hydrogen and hydrocarbons that may appear in the gases.

As dry air is composed of 20.99 per cent of oxygen by volume the complete combustion of one pound of carbon with the exact theoretical amount of air would produce a flue gas whose carbon dioxide content would be 20.99 per cent, and whose nitrogen content would be the same as that of dry air. Oxygen and carbon monoxide would not appear in

the flue gases. If the carbon is incompletely oxidized, carbon monoxide will appear in the analysis. If an excess of air is supplied, the gases will have a free oxygen content.

Practically all fuels contain hydrogen and hydrocarbons. The combustion of the hydrogen produces water vapor which does not appear in the dry gases when analyzed. The oxygen for the production of the water vapor is obtained from air, which leaves a nitrogen residue and which increases the percentage of nitrogen in the flue gases. Also, the amount of nitrogen contained in the fuel has the same effect. Coals which contain hydrogen and nitrogen, therefore, give flue gases which have nitrogen contents of 81 to 85 per cent. The sum total of the other constituents of flue gas in an actual analysis may range from 15 to 19 per cent.

Flue-gas analyses should be made of representative gas samples. The sample should be drawn from the furnace or boiler setting through a single open-ended metal or fused quartz tube. During the time that the sample is being drawn, the open end of the tube should be placed in various locations in the region from which a gas sample is desired. Sampling-tube locations at bends and in pockets below dampers are to be avoided. When a sample is drawn from the furnace, or the first passes of a boiler, or any region of extremely high temperature, a fused quartz tube must be used. Metal sampling tubes are likely to be oxidized by the oxygen of the gas sample at high temperatures. The reaction results in a reduction of the oxygen content of the gas sample, and the data obtained will not be representative of the flue gases being discharged.

The gas sample drawn for analysis should be collected in a container, over mercury or water well saturated with the flue gases. The sample should be aspirated into the container by a strong, continuous, and uniform process. Some sort of a filter or separator for moisture, tar, and dirt is desirable in the line between the sampling tube and the gas container. The gas anaylsis should be made immediately after the sample is collected. In many cases it is preferable to draw the filtered gas directly from the sampling tube into the analyzing apparatus.

81. Flue Gas Analyzers. Many commercial power plants or boiler rooms are provided with equipment which indicates or records continuously the CO_2 content of the flue gases. Some CO_2 recorders or indicators are delicate and easily deranged and require much attention. The indications given by nearly all recorders lag more or less behind the flue-gas production. All CO_2 recorders should be checked frequently for accuracy. The CO_2 content alone is not an indication of efficient combustion, as high percentages of CO_2 may also be accompanied by appreciable percentages of CO, owing to the small percentage of excess

air being supplied. Therefore, more complete analyses are essential at frequent intervals.

A portable analyzer which may be used for test work and for checking CO_2 recorders and indicators is either the Orsat apparatus or some of its modifications. A typical Orsat apparatus is shown by Fig. 26. The essential parts of the Orsat are the measuring burette *A*, the leveling bottle *F*, the gas cleaner *H*, and the absorption pipettes *B*, *C*, *D*, and *E*. The measuring burette *A*, the filter *H*, and the absorption pipettes are interconnected by means of a manifold having accurately ground cocks *I*. The measuring burette *A* is water-jacketed to prevent temperature and density changes of the gas while the analysis is in progress. The filter *H* may be filled with spun glass or similar material.

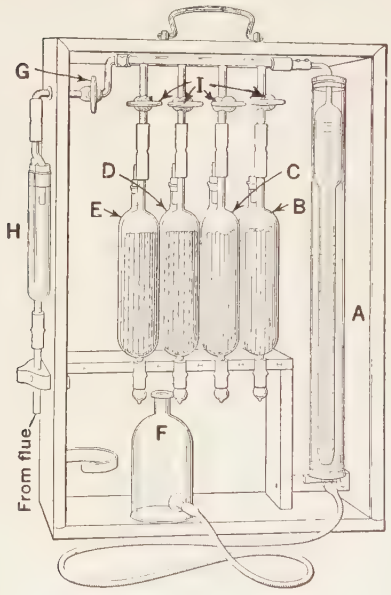


FIG. 26.—Orsat apparatus.

Pipette *B* is partially filled with a 40 per cent solution of caustic potash, KOH. This reagent absorbs CO_2 and is the first one used during an analysis. Pipette *C* contains an alkaline solution of pyrogallic acid. The pyrogallol absorbs oxygen and is made up of a mixture of equal volumes of 33 per cent KOH and 25 per cent pyrogallic acid solutions. The two pipettes *D* and *E* contain an acid solution of cuprous chloride which is used for the absorption of carbon monoxide, CO. Pipettes *D* and *E* should contain some fine metallic copper to keep the solution energized.

Each absorption pipette contains a number of small glass tubes, which provide a greater amount of wet surface, to facilitate the gas absorption during the time that the flue gas is held in the pipette. The flue gas during analysis is successively passed through pipettes *B*, *C*, and *D* or *E*. The analysis must be made in this order, and the absorption of each constituent must be complete before the gas is exposed to the next reagent. The nitrogen content, N_2, of the flue gas is obtained by subtracting from 100 the sum of the percentages of CO_2, O_2, and CO. The reagents in the various pipettes should be fresh and should be protected from the air when not in use. All connections between the various parts of the apparatus must be gas tight. Frequent tests for leakage from the apparatus are essential.

82. Actual Weight of Combustion Air.—The analysis of the flue gases formed, the ultimate analysis of the fuel burned, and the molecular weights of the principal constituents of the flue gases provide data for calculating the actual amount of air used per pound of fuel. The percentages of CO_2, O_2, CO, and N_2 of the flue gases are to be taken by volume, while the carbon content in the fuel is by weight.

If 100 mol-volumes of flue gases are considered, the percentages of the flue-gas analysis represent the mol-volumes of each constituent present (Art. 74). The weight of nitrogen is $28 \times N_2$, and the weight of carbon appearing as CO_2 and CO is $12(CO_2 + CO)$. Therefore, the weight of nitrogen per pound of carbon is

$$\frac{28N_2}{12(CO_2 + CO)}.$$

As one pound of air contains 0.769 lb. of nitrogen, the air, W_c, per pound of carbon, is

$$W_c = \frac{28N_2}{12(CO_2 + CO) \times 0.769}. \tag{70}$$

If the carbon content C_f of the fuel were completely burned, the actual air per pound of fuel would be

$$W_{cf} = W_c \times C_f$$

or

$$W_{cf} = \frac{28N_2}{12(CO_2 + CO) \times 0.769} \times C_f. \tag{71}$$

There is likelihood of unburned carbon appearing in the ash and refuse from solid fuels. Under such conditions, calculations involving the use of flue gas analyses should be based on the carbon actually burned per pound of fuel. The actual carbon consumed per pound is

$$C_1 = \frac{W_f C_f - W_r C_r}{W_f \times 100}. \tag{72}$$

C_1 = carbon burned per lb. of fuel, lb.
W_f = weight of fuel fired, lb.
C_f = carbon of the fuel from ultimate analysis, per cent.
W_r = weight of ash and refuse from W_f lb. of fuel, lb.
C_r = carbon content of the ash and refuse, per cent.

Equation 71 may then be written

$$W_{aa} = \left[\frac{28N_2}{12(CO_2 + CO) \times 0.769}\right] \times \left[\frac{W_f C_f - W_r C_r}{W_f \times 100}\right]. \tag{73}$$

W_{aa} = the actual weight of dry air used per lb. of fuel fired, lb.

When account is taken of the nitrogen of coal, its percentage should be divided by 76.9, and the weight of air equivalent should be deducted from W_{aa}. This correction represents the air equivalent of the nitrogen content of the flue gases produced by the presence of nitrogen in the fuel.

83. Weight of Excess Air.—Owing to furnace and burner limitations, more than the theoretical amount of air is always required for the combustion of a fuel. The difference in amount of air actually used per pound of fuel and that theoretically required is the excess air used. Excess air is usually expressed as a percentage:

$$E_a = \frac{W_{aa} - W_{ta}}{W_{ta}} \times 100. \tag{74}$$

E_a = percentage of excess air.
W_{aa} = actual air used per lb. of fuel, lb.
W_{ta} = theoretical air required per lb. of fuel, lb.

The percentage of excess air required for good combustion may range from 10 to 60. Often in burning solid fuels, holes in the fire, thin fires, strong draft, or leaky boiler and furnace settings may cause the actual percentage of excess air to be large. Improper adjustments of oil or gas burners may allow a great volume of air to pass through the furnace.

Excess air above that absolutely essential to good combustion always represents a distinct and often a large amount of heat lost in the chimney gases. The extra air dilutes the flue gases and lowers their temperature; so that less heat is transferred to the water in the boiler. Because of the increased weight of flue gases formed, the amount of heat carried away is increased.

84. Weight of Flue Gas Per Pound of Fuel.—On the basis of reasoning identical with that used to express the actual air used per pound of fuel, the weight of the dry gases formed per pound of carbon is

$$W_g = \frac{44CO_2 + 32O_2 + 28(CO + N_2)}{12(CO_2 + CO)}. \tag{75}$$

The symbols CO_2, O_2, CO, and N_2 all represent percentages by volume as determined by the flue gas analysis.

Equation 75 may be simplified as

$$W_g = \frac{44CO_2 + 32O_2 + 28[100 - (CO_2 + O_2 + N_2) + N_2]}{12(CO_2 + CO)}.$$

$$= \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}.$$

The actual dry gas formed per pound of fuel is found by $C_1 \times W_g$.

$$W_{dg} = \left[\frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}\right]\left[\frac{W_f C_f - W_r C_r}{W_f \times 100}\right]. \quad (76)$$

The actual weight of dry air supplied per pound of fuel as burned may be obtained by use of the actual weight of dry gases W_{dg} produced, as calculated by use of equation 76. Such a calculation involves the deduction of the weight of carbon C_1 actually burned per pound of fuel and the addition of the oxygen necessary for the combustion of the hydrogen. The nitrogen from the air used for the burning of the hydrogen appears in the total nitrogen content of the flue gases.

$$W_{aa} = W_{dg} - C_1 + 8\left(H - \frac{O}{8}\right). \quad (77)$$

Calculations made by use of equation 77 will approximately check those made by use of equation 73.

85. Heat Loss Due to the Formation of Carbon Monoxide.—The formation of carbon monoxide represents a distinct heat loss occasioned by incomplete combustion. This loss per pound of fuel is

$$h_{CO} = 10{,}160 \times \frac{CO}{(CO_2 + CO)}\left[\frac{W_f C_f - W_r C_r}{W_f \times 100}\right]. \quad (78)$$

PROBLEMS

1. Calculate the theoretical amount of air required for the combustion of a Williamson County, Illinois, bituminous coal.

2. Calculate the theoretical amount of air necessary for the complete combustion of a Cambria County, Pennsylvania, bituminous coal.

3. Calculate the volume of the theoretical air per pound of Kanawha County, West Virginia, coal. Air temperature 70 deg. F. Barometer 29.3 in. of mercury.

4. Calculate the volume of the theoretical air per pound of Green County, Indiana, coal. Air temperature 60 deg. F. Barometer 29.6 in. of mercury.

5. Calculate the theoretical amount of air, in cubic feet, necessary for the combustion of one cubic foot of blast-furnace gas of the following analysis: H_2, 3.2; CO, 26.2; CO_2, 13.0; and N_2, 57.6 per cent.

6. Calculate the theoretical amount of air in cubic feet necessary for the combustion of 1000 cu. ft. of coke-oven gas of the following analysis: H_2, 57.4; CH_4, 28.5; C_2H_6, 2.9; CO, 5.1; CO_2, 1.4; O_2, 0.5; and N_2, 4.2 per cent.

7. During a boiler test, 20,000 lb. of Sangamon County, Illinois, bituminous coal were fired. Ash and refuse collected amounted to 3180 lb. The carbon content of the ash was 15 per cent. The average flue-gas analysis percentages were CO_2, 11.5; O_2, 6.5; CO, 0; N_2, 82. Find the actual weight of air used per pound of fuel as burned.

8. During a boiler test, 18,690 lb. of Hocking County, Ohio, coal were fired. The ash and refuse collected weighed 1065 lb. and the carbon content of the ash and refuse was 25 per cent. The flue-gas analysis gave the following average per-

centages: CO_2, 14.3; O_2, 3.6; CO, 0.3; and N_2, 81.8. Find the actual weight of air used per pound of coal as burned.

9. A coal containing 65 per cent carbon required, theoretically, 9.45 lb. of air per pound of coal for combustion. During combustion, 3 per cent of the carbon fired was lost as unburned carbon in the ash and refuse. If the flue-gas analysis obtained was CO_2, 9.5; O_2, 9.0; CO, 0; and N_2, 81.5 per cent, find the percentage of excess air used.

10. Find the percentage of excess air used in problem 7.

11. Find the weight of dry gases formed in problem 7.

12. Find the weight of dry gases formed in problem 8.

13. What is the heat loss due to the formation of carbon monoxide if Crawford County, Kansas, coal is burned with a carbon loss in the ashes of 0.03 lb. per lb. of coal to give a flue-gas analysis of CO_2, 15.5; O_2, 1.0; CO, 1.6; and N_2, 81.9 per cent?

14. What was the heat loss due to the formation of carbon monoxide in problem 8?

15. A test of a small stoker-fired boiler gave the following combustion data: total coal fired, 610.5 lb.; total ash and refuse, 62.5 lb. Analysis of the coal as fired, percentages by weight, C, 59.58; H, 4.53; O, 8.58; N, 2.03; S, 4.35; moisture, 8.70; and ash, 12.23. Analysis of the ash and refuse, percentages by weight, moisture, 0.07; volatile matter, 0.0; fixed carbon, 4.46; and earthy matter, 95.47. Analysis of the dry flue gases, percentages by volume, CO_2, 11.1; O_2, 8,6; CO, 0.0; and N_2, 80.3. Heating value of the coal as fired, 11,036 B.t.u. per lb.; room temperature, 81 deg. F.; temperature of the escaping flue gases, 432 deg. F. Find: (*a*) The percentage of excess air used; (*b*) the weight of dry gases formed per pound of coal as burned; and (*c*) the percentage of the heat of the fuel lost as sensible heat in the dry flue gases. Mean specific heat of the flue gases, 0.25.

16. Coal of the following analysis, percentages by weight, was used in a boiler furnace: C, 78.38; H, 4.69; O, 6.14; N, 1.43; S, 0.67; moisture, 2.86; and ash, 5.83; heating value, 14,105 B.t.u. per lb. as fired. The volumetric analysis of the dry flue gases was: CO_2, 15.3; O_2, 1.5; CO, 2.1; and N_2, 81.1 per cent. The carbon loss in the ash and refuse was 0.15 lb. per pound of coal fired. The boiler room temperature was 85 deg F. and that of the flue gases in the breeching was 650 deg. F. Mean specific heat of the flue gases, 0.24. Find: (*a*) the total weight of air used per pound of fuel; (*b*) the percentage of the heat of the fuel lost due to the formation of CO; and (*c*) the percentage of the heating value of the fuel lost as sensible heat in the dry flue gases.

17. The ultimate analysis of a coal was: C, 51.45; H, 4.02; O_2, 7.28; N, 0.93; S, 3.92; ash, 16.69; and moisture, 15.71 per cent. A weight of 15,395 lb. of the coal was burned and 2816 lb. of ash and refuse recovered. Carbon content of the ash and refuse, 20.9 per cent. If 13.3 lb. of dry gases were formed per pound of coal as burned, how much air was supplied per pound of coal?

CHAPTER VII

STEAM GENERATORS OR BOILERS

86. Development.—*A steam-generating unit*, as defined by the American Society of Mechanical Engineers, is a combination of apparatus for producing, furnishing, or recovering heat together with apparatus for transferring the heat so made available to the fluid being heated and vaporized. The unit may include any or all of the following component parts: boiler, furnace, fuel-firing equipment, water walls, water floor, water screen, superheater, reheater, economizer, and air heater. The word " boiler " refers to the physical change of the contained fluid and not to the purpose of the heating process; hence, engineers are encouraging the use of the term " steam generator " in place of " boiler " as the name for the primary part of a steam-generating unit.

Fifty years ago, a steam pressure of 100 lb. per sq. in. was considered dangerously high. Twenty years ago, high pressure referred to anything above 150 lb.; today (1933), pressures under 400 lb. per sq. in. are considered as ordinary. Boilers operating at the critical pressure of water vapor, about 3200 lb. per sq. in., have been used successfully for several years in a number of European installations.[1] In the United States, the trend toward higher pressures and temperatures both in central stations and in industrial plants is firmly established, as evidenced by numerous 1400-lb. installations, one of 1800 lb. in an industrial plant, a 2500-lb. testing boiler, and several experimental steam generators [2] for pressures up to 5000 lb. per sq. in. Single units are now in operation capable of generating over a million pounds of steam per hour, and boiler outputs of more than two million pounds of steam per hour seem quite possible of attainment in the near future; steam temperatures of 850 deg. F. are not uncommon, and the general

[1] "High-Pressure Boilers in Central Europe," by A. Weisselberg; *Power*, October 14, 1930, p. 600, and Oct. 21, 1930, p. 644.

"High-Pressure Steam in Germany," by W. Parey; *Power Plant Engineering*, Nov. 1, 1930, p. 1271.

"Benson Boiler of 7300 Hp."; *Power Plant Engineering*, Nov. 15, 1931, p. 1097.

[2] "Once-Through Series Boiler for 1500- to 5000-Lb. Pressure," by H. J. Kerr, paper RP–54–1a; "Characteristics of a High-Pressure Series Steam Generator," by Potter, Solberg, and Hawkins, paper RP–54–1b; A.S.M.E. Annual Meeting, December, 1931.

interest in increasing steam temperatures continues with several units now operating at 1000 deg. F. Experiments have demonstrated the workability and reliability of high-pressure steam-generating equipment and have opened the way for further improvements. With materials under development for high temperatures as well, it seems reasonable to expect commercial steam generators to be built for 2500 lb. per sq. in. pressure and 1000 deg. F. steam temperature.

Mercury may be vaporized in a special boiler at temperatures higher than those generally practicable with steam. A description of the mercury-steam cycle is given in Art. 257, Chapter XV.

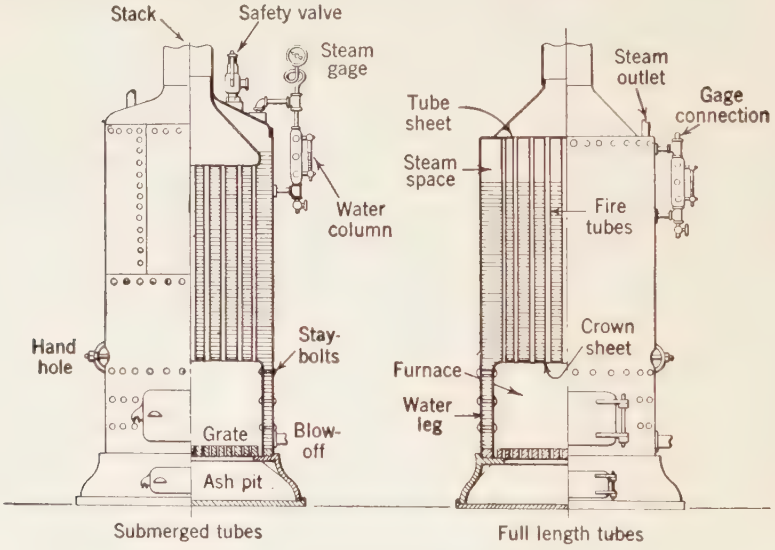


Fig. 27.—Vertical fire-tube boilers.

87. Classification.—Steam boilers may be classified according to: (1) relative position of the water and hot gases, as *water-tube* and *fire-tube*; (2) arrangement of tubes, as *vertical*, *horizontal*, and *inclined*; (3) curvature of tubes, straight-tube and bent-tube; (4) location of furnace, *externally fired* and *internally fired*; and (5) nature of service, *stationary*, *locomotive*, and *marine*.

The selection of a steam boiler for a particular service may depend on a number of factors, economical and practical, which are beyond the scope of this book. In the following articles the engineering features of several popular types of boilers will be described in detail.

88. Vertical Fire-tube Boilers.—In the fire-tube boiler the hot gases from the fire pass through the tubes. Figure 27 shows two types of vertical fire-tube boilers, having internal furnaces and with tubes

extending from the crown sheet to a tube sheet at the top of the boiler. They are built in standard sizes ranging from 20 to 48 in. in diameter and from 48 to 120 in. in length, with corresponding ratings of 3 to 50 hp. The tubes are usually 2 in. in diameter for anthracite coals and 3 in. in diameter for bituminous coals; the working pressure seldom exceeds 100 lb. per sq. in. gage.

This type of boiler is generally used where the chief requirements are: low steam pressures, compactness, portability, low first cost, rapid steaming ability, and small quantity of steam (100 to 1500 lb. per hour). Some of the disadvantages are: small steam capacity, poor economy because of high temperature of gases entering the stack, inaccessibility for inspection and cleaning, and fluctuation of steam pressure with varying demands for steam.

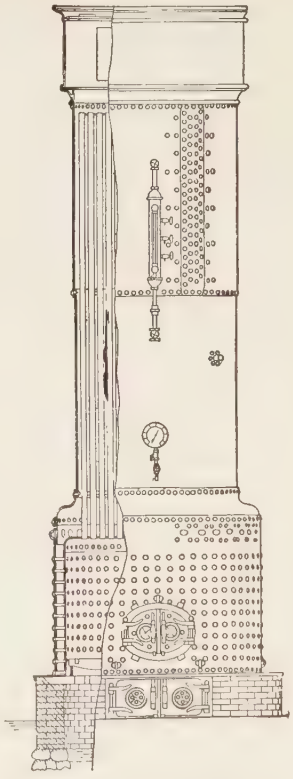


FIG. 28.—Manning boiler.

Other types of vertical fire-tube boilers, of which the Manning (Fig. 28) is a well-known example, are not limited to small sizes and have been built with a heating surface [3] of 6000 sq. ft. per unit. The enlarged and strongly stayed circular firebox rests on a permanent brick or concrete setting which forms the ashpit. Some of the disadvantages of the smaller types are minimized, so that the effects are: (*a*) improved economy because of increased steam space, larger grate area, and longer tubes; (*b*) suitability for steam pressures up to 200 lb. per sq. in.; and (*c*) a small floor space is required as compared to horizontal types of equal capacity.

89. Horizontal Return-tubular Boilers.—In Fig. 29 is shown a longitudinal view of a horizontal return-tubular (H.R.T.) boiler and setting. The furnace consists of the combination of grates, bridge wall, fire and ash doors, the ashpit, and the combustion space. The front and sides of the furnace are lined with firebrick. The boiler shell is supported by steel beams and hangers; lugs resting on brickwork may be used for small horizontal return-tubular boilers.

[3] *Boiler heating surface* consists of that portion of the surface in contact with the water being heated on one side and the gas or refractory being cooled on the other; this surface is measured on the gas side.

The hot gases pass over the bridge wall, along the bottom of the boiler shell, and then return through the fire tubes, toward the front of the boiler. From the front end of the tubes the products of combustion pass through the breeching into the stack.

Standard sizes of these boilers are built, in accordance with the A.S.M.E. Boiler Code, from 36 in. in diameter and 8 ft. long, to 84 in. in diameter and 20 ft. in length; ranging from 150 to 3500 sq. ft. of heating surface and steam pressures up to 175 lb. The tubes are usually 3, $3\frac{1}{2}$, or 4 in. in diameter, the smaller tubes for low-volatile fuels, and the larger tubes for high-volatile fuels. Return-tubular boilers are

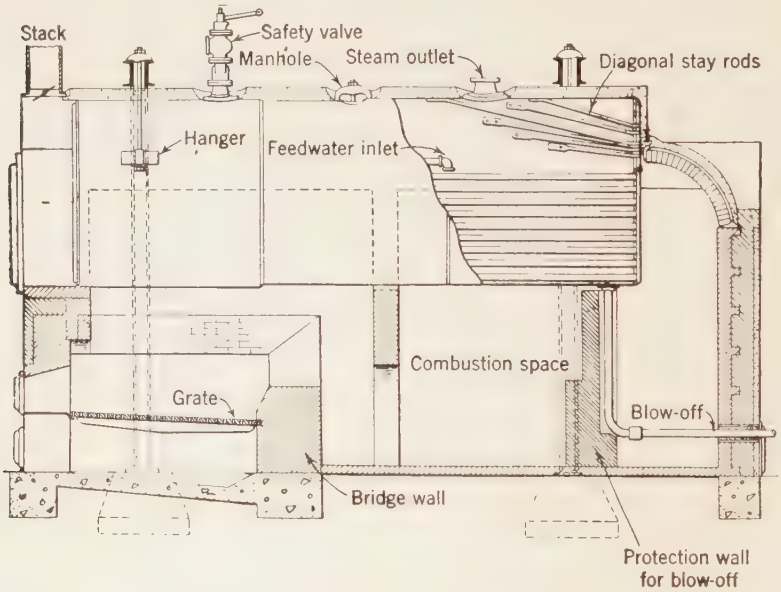


Fig. 29.—Horizontal return-tubular boiler.

widely used in small industrial plants because of their compactness, low first cost, ease of cleaning, and water-storage capacity. They have, to some extent, the disadvantages of the vertical-tubular boilers.

A return-tubular firebox boiler of the portable type, as illustrated by Fig. 30, is being used frequently where a compact moderate-capacity and self-contained unit is desired. The front of the boiler is cylindrical and extends over the furnace; the rear end is oval, the lower portion extending below the cylindrical part far enough to provide for the short direct tubes leading from the furnace to the back connection. The hot gases pass through the direct tubes into the back connection, and then return through the return tubes into the front end of the upper section of the boiler to the stack. The front and sides of the

furnace are lined with firebrick. This type of boiler is available in standard sizes ranging from 200 to 1500 sq. ft. of heating surface and is intended for pressures not exceeding 125 lb. per sq. in.

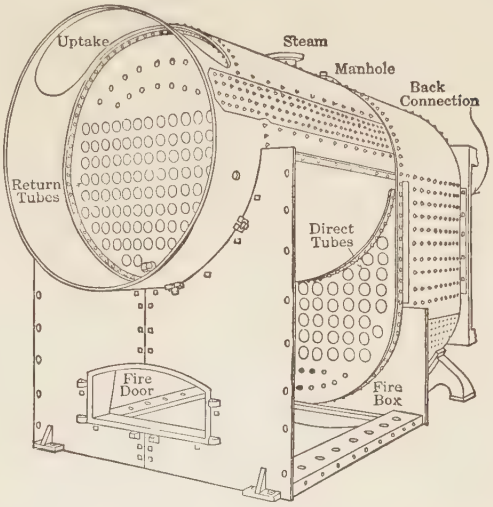


FIG. 30.—Horizontal return-tubular portable fire-box boiler.

90. Horizontal Water-tube Boilers: Longitudinal Drum.—Figure 31 shows a longitudinal-drum Babcock and Wilcox horizontal water-tube boiler, typical of this class. This particular boiler is made in units from 750 to 12 000 sq. ft. of heating surface, and for steam pressures as high as 500 lb. per sq. in. Distinguishing features of this type are: hori-

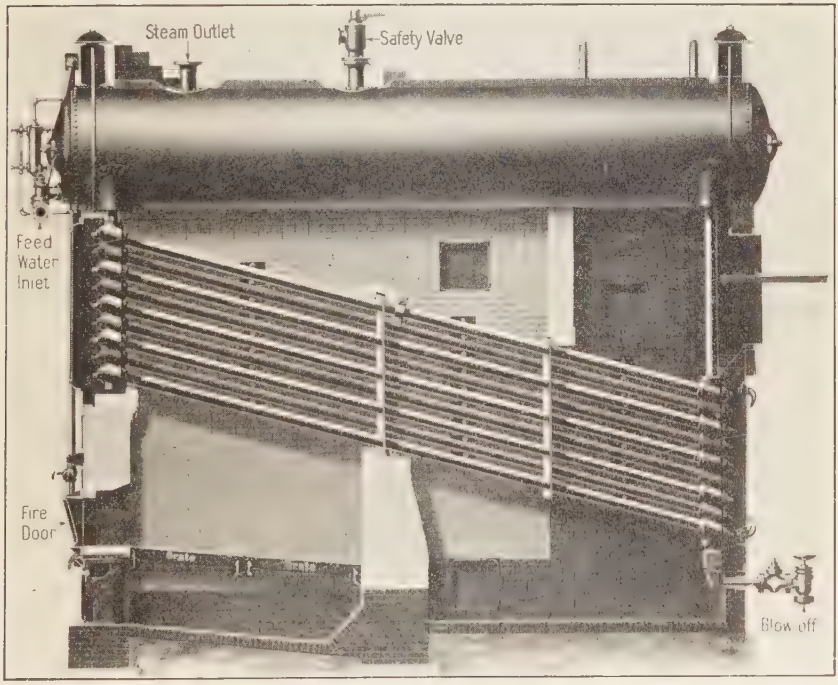


FIG. 31.—Babcock and Wilcox longitudinal-drum boiler with vertical headers.

zontal drum or drums, vertical sectional headers, straight tubes inclined at an angle of 22 deg., and vertical (or nearly so) flue-gas baffles. The tubes, usually 4 in. in diameter and 18 to 20 ft. in length, are arranged in vertical and horizontal rows and are expanded into cast-iron or pressed-steel headers. Two vertical rows of tubes are fitted to each header, the tubes being "staggered" with respect to each

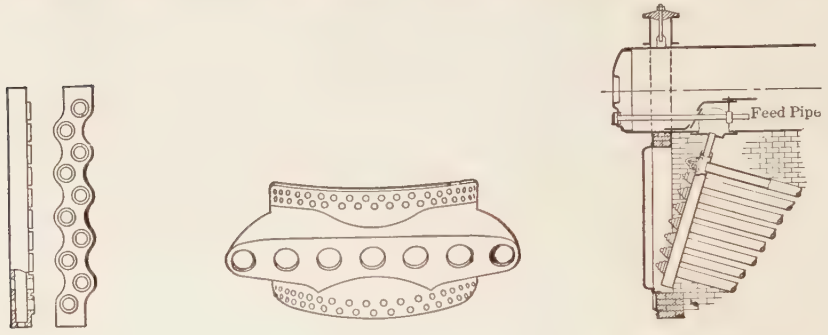


FIG. 32.—Details of B. and W. boiler construction.

other, as shown in Fig. 32. The headers are connected with the steam drum by short tubes expanded into a cross box, Fig. 32, which in turn is riveted to the drum as shown in Fig. 31. The headers are either vertical, Fig. 31, or inclined, Fig. 32 (right).

The boiler is suspended by stirrups from beams resting on suitable steel columns; thus the entire structure is free to expand or contract without injury to boiler or brick setting; also this makes it possible to repair the brickwork without disturbing the boiler or its piping.

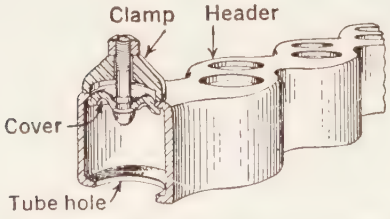


FIG. 33.—Hand-hole details, B. and W. boiler.

The feedwater entering the front end of the drum, Fig. 32, is readily picked up by the water circulating in the boiler, and carried toward and down the rear header; circulation is effected by the difference in density between the column of water in the rear headers, and the mixed steam and water in the front headers. A deflecting steam baffle, Fig. 32, in the drum over the front headers, prevents an excess of water from being carried into the steam space. The steam passes into a perforated pipe (dry pipe) at the top of the drum and out into the steam main.

Most of the impurities in the boiler water separate out and settle in the mud drum at the bottom of the rear headers, shown in Fig. 31.

These impurities are removed at intervals by opening the valves in blow-off pipes leading from the mud drum. Hand holes, Figs. 32 and 33, are provided opposite the tube holes in each header, for the purpose of assembly and cleaning or replacing tubes. The surfaces surrounding the holes are smoothly machined to form gasket joints with the hand-hole covers. Scale is removed from the inside of the drum by the use of hammer and chisel, workmen entering through the man holes. Soot and ash, which accumulate on the outer surfaces of the tubes, are

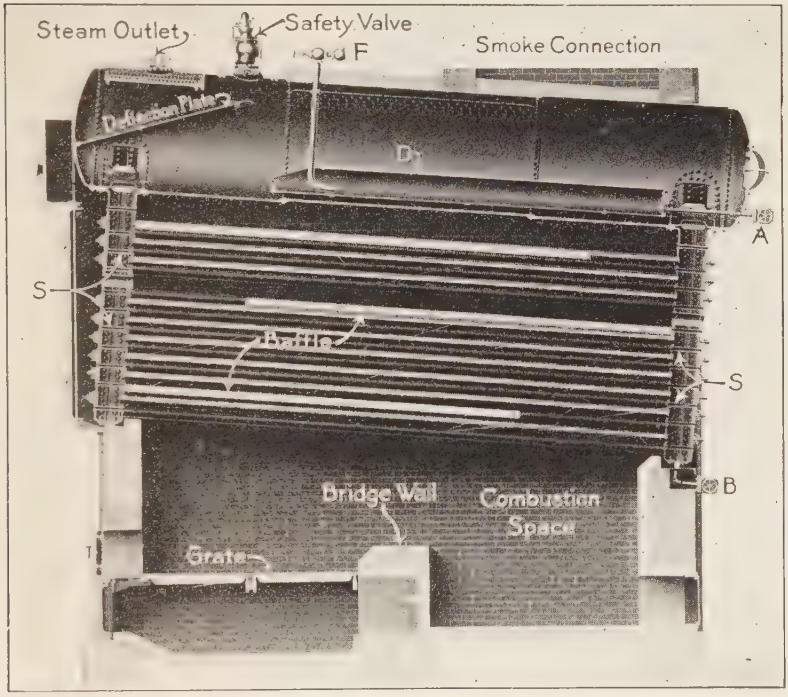


FIG. 34. — Heine H-type, longitudinal-drum boiler, horizontally baffled.

removed periodically by hand brush or scraper, or by a steam or air jet; the necessary brush, scraper, or nozzle is inserted through openings in the side walls of the brickwork.

Baffle plates separate the flue-gas areas around the tubes into compartments called passes. The baffles are so located that the gas-passage areas are ample, and so proportioned as to give a maximum heat absorption from the gases without undue frictional resistance. In Fig. 31 the front baffle, which is made of cast iron and firebrick, is supported by the bridge wall and the tubes above the bridge wall, while the rear baffle is supported by a steel beam and the tubes. The products of

combustion travel upward through the first pass across the tubes to a space beneath the drum. The hot gases are turned downward over the front baffle, across the central portion of the tubes through the second pass, and then around the lower end of the rear baffle into the third pass, and across the lower ends of the tubes to the damper and stack connection.

A general assembly of a Heine water-tube boiler is shown by Fig. 34. It consists essentially of an inclined drum, two sheet-steel headers, inclined straight tubes connecting these headers, and horizontal baffles. A throat opening is cut into the bottom of the drum near each end, for the header (water leg) connections, see Fig. 35.

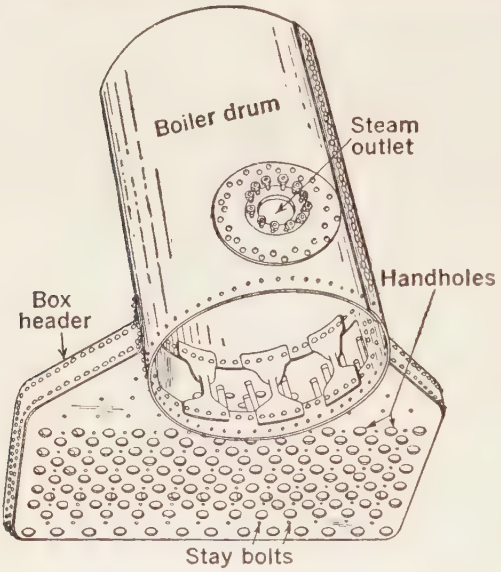


Fig. 35.—Header and drum details, Heine H-type boiler.

The feedwater enters at *F*, Fig. 34, at the top and near the front end of the drum, and goes into a steel mud drum *D*; the latter is supported several inches above the bottom of the boiler drum. The greater part of the impurities is deposited in the mud drum and removed periodically through a blow-off pipe *A*. A blow-off connection *B* is also shown at the bottom of the rear header. The warmed water rises and flows out of the front end of the mud drum *D*, and there joins the circulation of the boiler water.

The products of combustion are generally directed along the tubes by longitudinal baffles, and thence along the under side of the drum to the smoke connection. Scale, which accumulates in the tubes, is removed by mechanical tube cleaners which are inserted through the hand holes in the headers opposite the ends of the tubes. Soot and dust may be blown from the tubes by steam lance pipes inserted through the hollow stay bolts *S* in the headers.

91. Horizontal Water-tube Boilers: Cross Drum.—The Walsh-Weidner boiler, Fig. 36, is typical of this class. Boilers of this type were originally designed to meet certain conditions of headroom, as in marine service; in recent years they have become popular for central

power stations and are particularly adaptable for large units (up to 50,000 sq. ft.) with high steam pressures and temperatures with an economy of floor space.

The drum is located across and above the tubes, directly above the rear header or between the tops of the headers. Special features of this boiler are: sectional sinuous steel headers, vertically baffled,

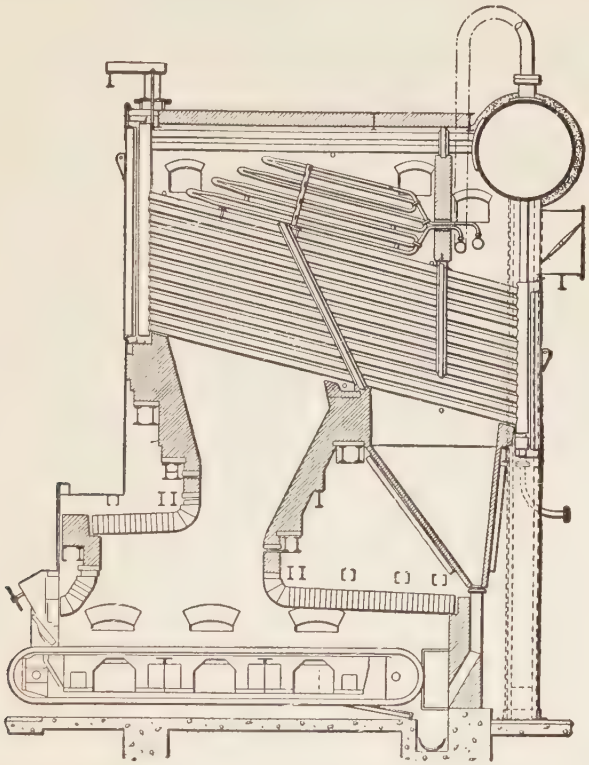


FIG. 36.—Walsh-Weidner cross-drum boiler with superheater, fired with chain grate stroker.

straight tubes 4 in. in diameter, unrestricted water circulation, very compact, tubes staggered vertically, and no stay bolts.

Figures 68, 69, 70, 71, and 85 show cross-drum boilers with some differences in design and arrangement of heating surfaces from the boiler illustrated in Fig. 36.

92. Vertical Water-tube Boilers: Straight-tube Type.—The Bigelow-Hornsby boiler, Fig. 37, is made up of cylindrical units, each unit consisting of a lower and an upper drum connected by straight tubes. Four units, as shown in Fig. 37, constitute a section, and any

number of sections may be connected together. The two front units are inclined over the furnace at an angle of about 68 deg. with the horizontal; the two rear units are vertical. The upper drums are connected to a horizontal steam drum by bent tubes, as shown; the lower drums also have interconnecting tubes. Feedwater enters the top drum of the rear units; passes down the rear tubes and then up the tubes of

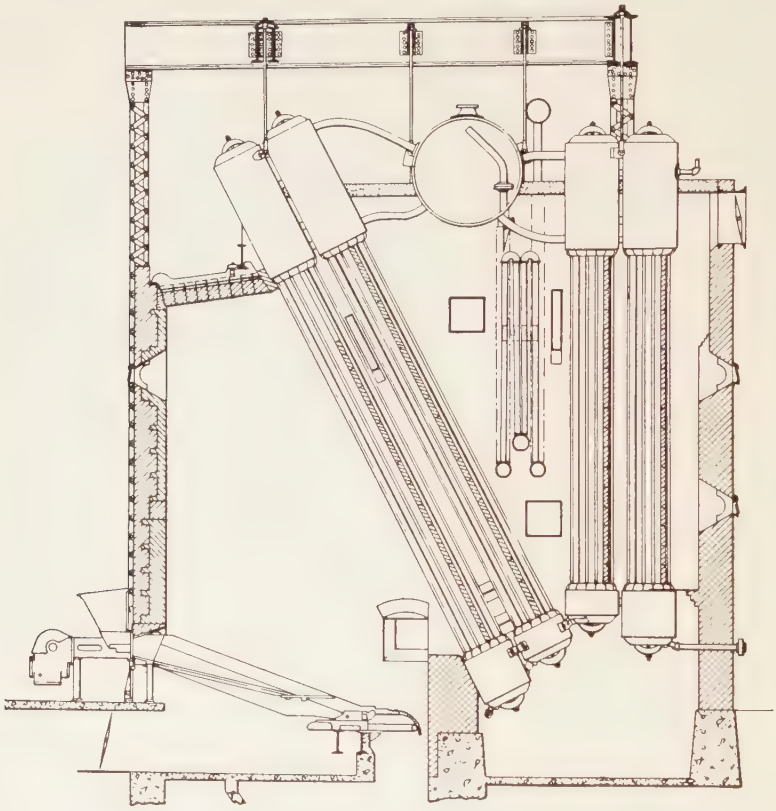


Fig. 37.—Bigelow-Hornsby boiler.

the front units. This boiler is built in sizes from 3000 to 25,000 sq. ft. of heating surface.

The arrangement of the baffling in the Bigelow-Hornsby boiler is unusual. Figure 38 shows a plan of the boiler units and baffles in the second, third, and fourth passes. These baffles extend along the entire length of the tubes between the units, and the gases pass over the tubes transversely. In the first pass the baffles do not extend to the top of the tubes; see Fig. 37. The baffles divide the products of combustion

into thin streams passing across the tubes, thus contributing to a more rapid heat transfer than is possible when there are large unrestricted

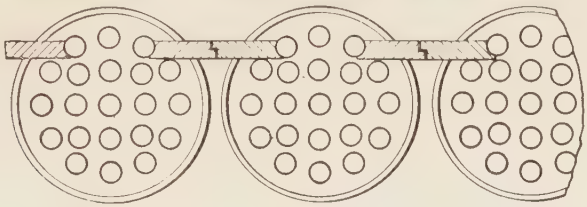


FIG. 38.—Plan of Bigelow-Hornsby boiler.

areas paralleled by the heating surface in the direction of flow of the gases.

The Wickes vertical water-tube boiler, Fig. 39, consists of an upper

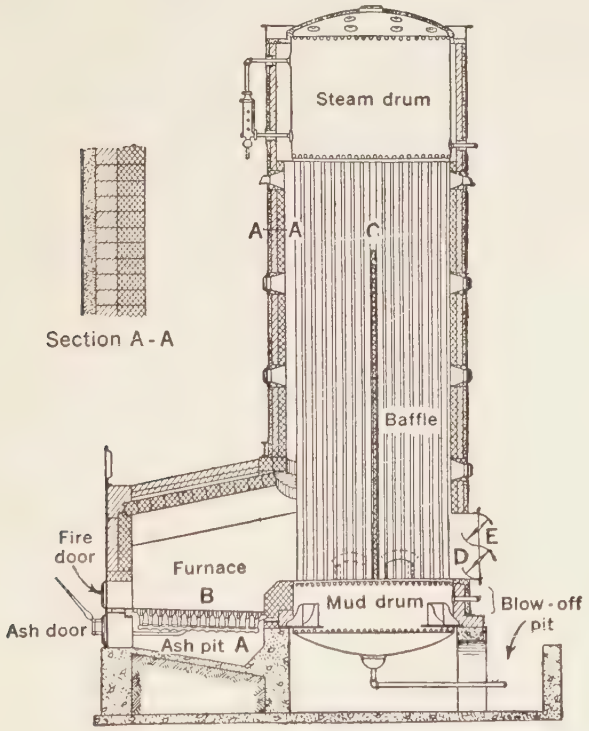


FIG. 39.—Wickes vertical water-tube boiler.

and a lower circular drum joined by straight vertical tubes. The tubes are arranged in parallel rows, and a vertical baffle *C* is placed in the center extending almost to the upper drum. The furnace is located in

a " Dutch oven," built out from the main boiler setting. The boiler is supported by four brackets riveted to the mud drum and resting upon plates fastened in the walls of the brickwork.

The steam drum has feed-pipe and water-column connections, also a man hole, and is normally one-third full of water; thus a large steam space and a large disengaging surface for steam separation are provided. The hot gases pass upward over the front tubes and downward over the rear tubes to the smoke connection *DE*. The feedwater enters at the rear near the bottom of the upper drum, and flows downward in the rear tubes and upward in the front tubes. The Wickes boiler is simple in design, requires small floor space, and is easy to inspect and clean. These boilers are built in varying sizes up to 5000 sq. ft. of heating surface and for steam pressures not exceeding 200 lb. per sq. in. gage.

93. Vertical Water-tube Boilers: Bent-tube Type.—Among the better-known designs of boilers that might be named under this heading are the Stirling, Badenhausen, Ladd, Erie City, Connelly, and Kidwell. These boilers have one or more horizontal steam drums above, connected by vertical or sloping tubes to one or more horizontal water drums below. This type of boiler has a large number of tubes, and therefore is suitable for high capacities.

One of the latest types of Stirling boilers is shown by Fig. 40. It consists of three transverse steam and water drums set parallel and connected to one mud drum by tubes so curved that their ends enter the drums at right angles to the surface. This curvature of the tubes gives ample and efficient provision for expansion and contraction. The front and middle steam drums are connected by curved equalizing tubes above the water line and curved circulating tubes below the water line, while the rear and middle drums are connected by curved equalizing tubes above the water line only.

The feedwater enters the rear upper drum and is delivered into a trough by which the water is distributed along the whole length of the drum. The water is gradually heated and partly purified as it passes downward to the mud drum, where the greater part of the material precipitated by the water remains until blown off. From the lower drum the water passes up the front bank of tubes; the steam formed passes to the central drum and then to the rear drum through the upper set of tubes. The water not evaporated passes to the central drum through the lower set of tubes. This water passes from the upper central drum down the middle bank of tubes to the mud drum and returns through the front bank of tubes. Steam is collected in a dry pipe at the top of the rear drum and passes from there to the superheater header. Safety valves are located on top of the rear drum and on the super-

heated steam header. The water column, located at one side of the front of the boiler, is connected to one end of the rear upper drum.

The design of this boiler produces ideal circulation of the water and

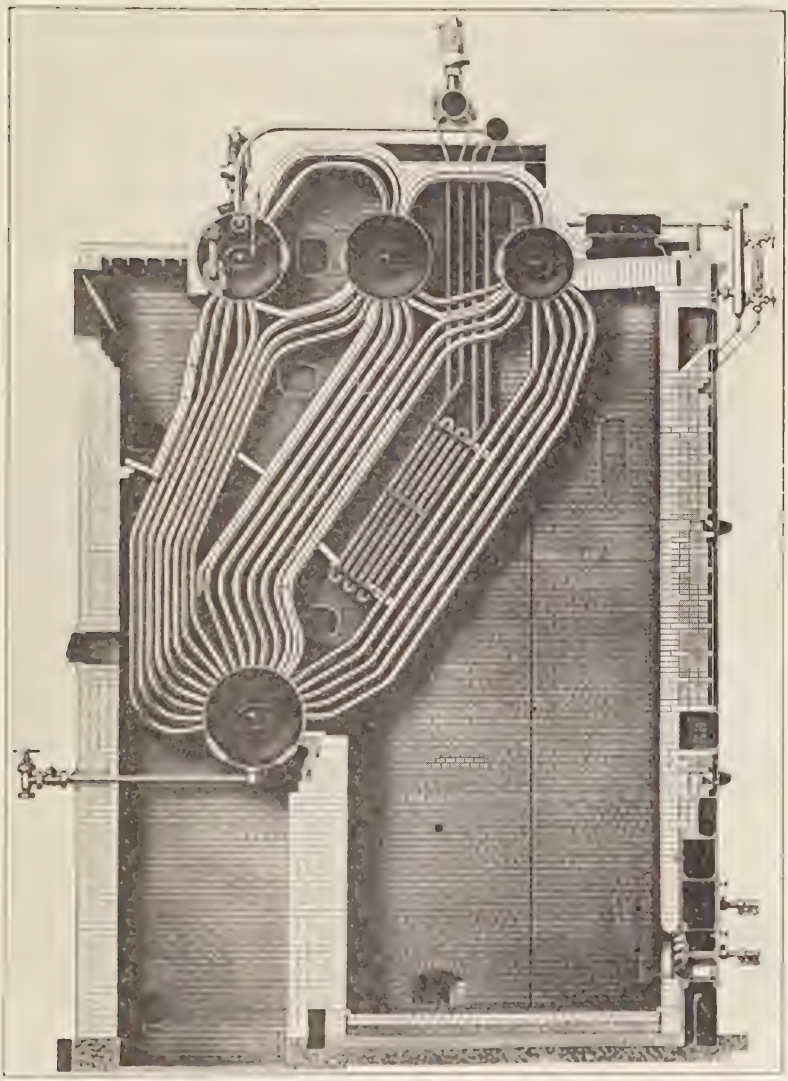


Fig. 40.—Stirling water-tube boiler with superheater and oil-burning furnace.

gases; that is, the coldest gases come in contact with the coldest water in the boiler, and the hottest gases come in contact with the hottest water. As the hot gases leave the burner, Fig. 40, they pass up between

the tubes and superheater to the front upper drum, then down around the tubes from the second drum, and again up between the tubes to

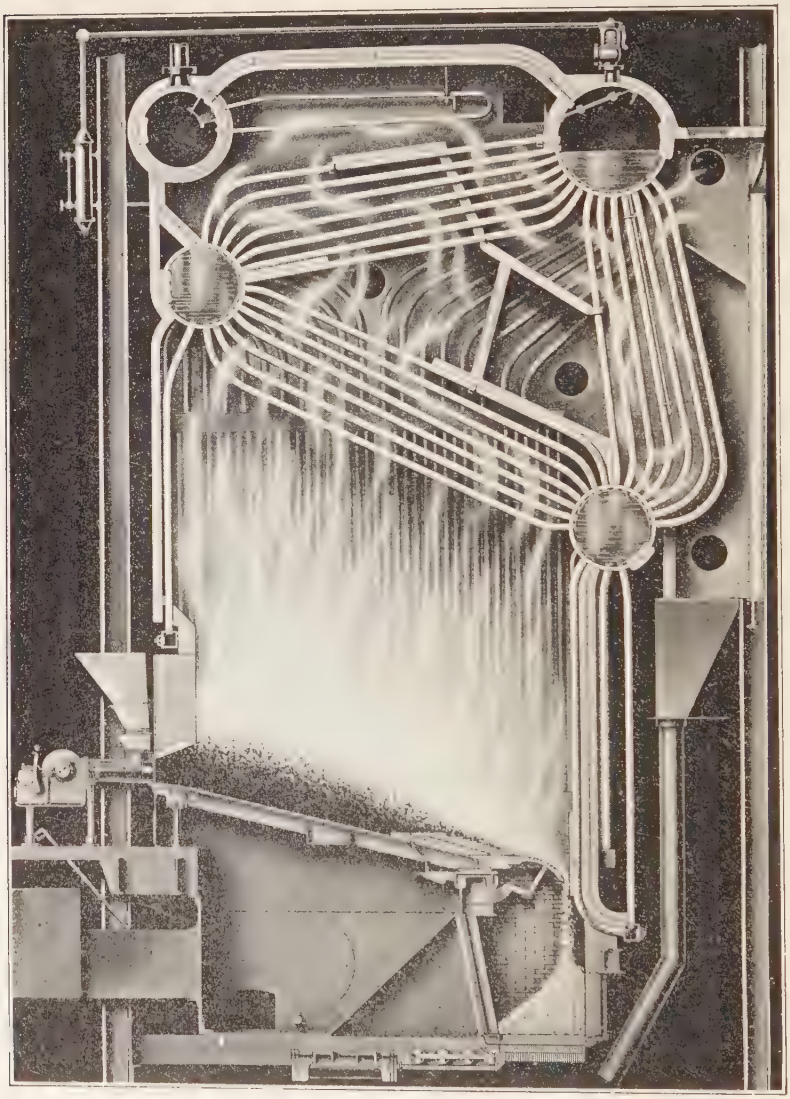


FIG. 41.—Riley-Badenhausen water-tube boiler with underfeed stoker and water-cooled furnace walls.

the rear upper drum. The burned gases leave the boiler at the rear, near the upper end of the last bank of tubes. The drums with their

connecting tubes are supported by a steel frame built into the brickwork of the boiler. The brick setting only serves to enclose the gases; it is under no strain due to the weight of the boiler.

The interiors of the drums are accessible, for inspection and cleaning, through man holes located in the drum ends. The tubes are cleaned by tube cleaners on the inside and by soot blowers on the outside. No hand holes or stay bolts are necessary. The chief disadvantage of bent-tube boilers is the initial cost and the necessity of carrying a large stock of replacement tubes, on account of their dissimilarity.

The Riley-Badenhausen boiler, Fig. 41, the Connelly, and the Kidwell boilers are similar in design. The area of each bank of tubes is practically the same; thus excellent circulation of water is insured, being unrestricted and continuous at all loads. The feedwater enters the top rear drum and passes down the rear bank of tubes to the mud drum. It flows up the front bank of inclined tubes to the lower front drum, and then across the upper bank of tubes to the top rear drum. These boilers have been called "ring flow," because of the circular path of the water through the tubes and drums as it is being heated. The steam is taken from the upper front drum; the upper connecting tubes serve as superheating surface.

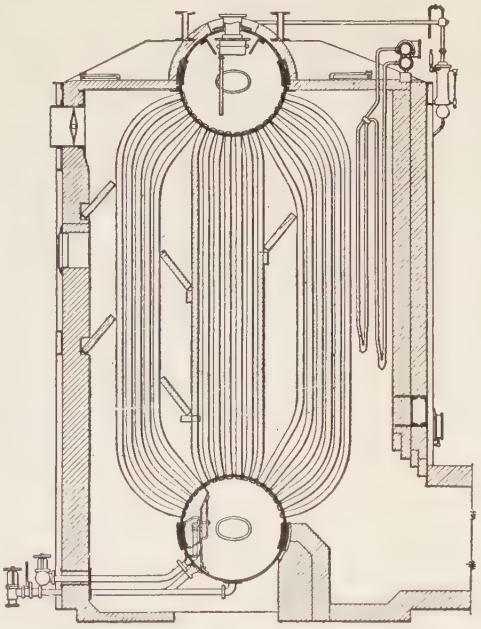


Fig. 42.—Small Ladd vertical boiler with superheater.

The small Ladd boiler, Fig. 42, the Casey-Hedges two-drum, and the Erie City vertical boilers in sizes up to 10,000 sq. ft. of heating surface have two drums and three parallel banks of tubes, as shown. In the Ladd boiler the feedwater enters a side chamber (feed box) in the lower drum and is directed upward through the rear bank of tubes, down the middle bank, and then passes upward in the front bank. The manufacturers claim that the use of this feed box in the lower drum causes the insoluble feedwater elements to be deposited near the blow-off, and prevents these particles from getting into the tubes to form scale.

There is an additional blow-off connection at the bottom of the lower drum.

The upper drum, Fig. 42, has the water level at the center line, thus affording a large steam-liberating surface. The lower drum is not sup-

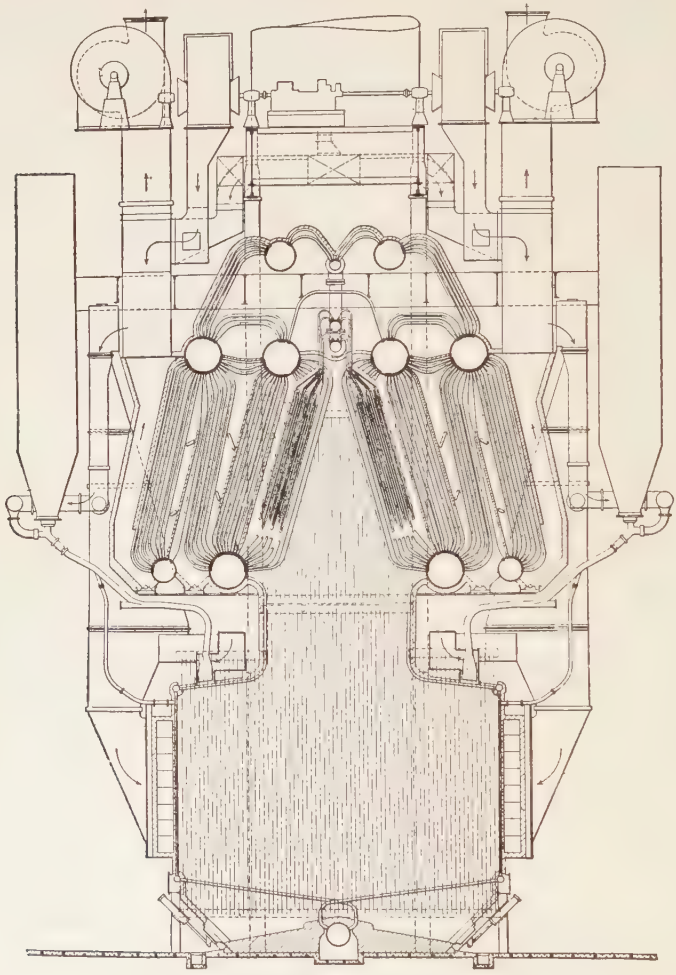


Fig. 43.—Large Ladd boiler with superheater, pulverized-coal burners, water screens, water-cooled furnace walls, air heaters, and integral economizers. Installed at East-River Station, New York City.

ported from below, but by the tubes which connect it to the upper drum; the curvature of the tubes insures ample facility for expansion and contraction. The shelf-like baffle extensions give a slightly transverse flow to the hot gases passing over the middle and last banks of

tubes. The vertical type of boiler is particularly adaptable to waste heat installations, it is easily forced, and it operates efficiently above its rated capacity.

The Ladd Double-Type boiler, Fig. 43, has been developed to meet requirements of heavy duty and sustained overloads. The unit illustrated has produced a million pounds of steam per hour for ten-hour periods at an average efficiency of 86.5 per cent; on peak loads it has operated at the rate of 1,270,000 lb. of steam per hour. This design combines economy of operation and low maintenance with unusual flexibility for adaptation to all combinations of size, type of fuel, pressure, capacity, superheat, and space requirements.

94. Boiler Capacity.—The capacity of a steam boiler is measured by the total heat transferred through the heating surfaces per hour. Inasmuch as this figure is numerically large, the *unit of evaporation*, which is equal to 1000 B.t.u., has been adopted by the American Society of Mechanical Engineers. Recently it was decided to call this unit the kilo B.t.u. (*kB*) and to use also 1,000,000 B.t.u. or the mega B.t.u. (*mB*) for large installations. Thus, the capacity of a boiler expressed in units of evaporation is the total heat transfer divided by 1000.

The capacities of boilers have hitherto been stated in terms of an irrational unit, the boiler horsepower. The boiler horsepower as originally standardized by the American Society of Mechanical Engineers in 1889 was based on a conventional engine steam rate of 30 lb. per horsepower per hour at 70 lb. per sq. in. gage pressure and feedwater at 100 deg. F. This corresponds to the evaporation of 34.5 lb. of water per hour from and at 212 deg. F. (33,472 B.t.u. per hour); which means the conversion of 34.5 lb. of water per hour *from* water at 212 deg. F. into dry and saturated steam *at* 212 deg. F. and, therefore, at atmospheric pressure (14.7 lb. per sq. in. abs.). Each pound of steam generated under these conditions requires the latent heat of vaporization at atmospheric pressure (14.7 lb. per sq. in. abs.), which is 970.2 B.t.u.

This weight and temperature of the water supplied to a boiler, and the pressure and quality of the steam leaving the boiler, can be obtained by test. As no power plant boilers receive feedwater at 212 deg. F. and convert it into steam at the same temperature, it is necessary to convert the weight of water actually evaporated per hour to the weight that would have been evaporated from and at 212 deg. F. and then divide by 34.5 to find the boiler horsepower developed.

At the present time, steam rates vary from 8 lb. per brake horsepower in large condensing turbines to 60 lb. for small non-condensing units, and the boiler horsepower has no connection whatever with the steam rate of the engine. It has never been used on the Continent or in England; in the United States, marine boilers are rated according to

the horsepower and steam rate of the engines they serve. Because the term boiler horsepower is still being used by some engineers it will be included in the following units of output for steam-generating units:

(*a*) *Actual evaporation*, pounds per hour, at observed steam pressure and quality or temperature, and at observed feedwater temperature.

Equivalent evaporation, pounds of water per hour which would have been evaporated from and at 212 deg. F. if the same amount of heat had been absorbed as under observed conditions.

Factor of evaporation, ratio of the heat absorbed by one pound of feedwater under observed conditions to that absorbed by one pound of water evaporated from and at 212 deg. F.

(*b*) Heat absorbed by water and steam in *kB* per hour,

$$Q = \frac{W_s(h - h_f)}{1000}. \tag{79}$$

(*c*) *Boiler hp.* $= \dfrac{W_s(h - h_f)}{970.2 \times 34.5}$, (80)

in which W_s = weight of feedwater supplied to the boiler, lb. per hour.
h = heat of one pound of steam at observed pressure and quality or temperature, B.t.u. per lb. above 32 deg. F.
h_f = heat of the liquid of one pound of feedwater at observed temperature, B.t.u. per lb. above 32 deg. F.

A term used especially with small boilers is the *rated horsepower* of a boiler; it is dependent upon the water-heating surface. The boiler heating surface consists of that portion of the surface in contact with the water being heated on one side and the gas or refractory being cooled on the other; this surface is measured on the gas side. All stationary boilers are rated on a basis of 10 sq. ft. of heating surface per boiler horsepower. By this method, boilers of equal heating surface are given the same rating, but one boiler may have its heating surface arranged more advantageously than another and thus have a greater capacity for like operating conditions. The rated horsepower does not express the limitations of capacity of present-day boilers, as most power-plant boilers can develop from 300 to 600 per cent of their normal rating and some operate for long periods at 250 per cent rating. The efficiency of the boiler may be decreased when it develops much more than its rated horsepower, see Fig. 44. This decrease in efficiency is generally not appreciable until the boiler is operating at 200 per cent or more of boiler rating, the decrease being due to the inability of the boiler to absorb the additional heat of the fire.

95. Boiler Performance. –The performance of a steam-generating unit may be expressed in terms of: efficiency, rate of combustion, evapo-

ration per pound of fuel, number of B.t.u. absorbed per square foot of heating surface per hour, and percentage of boiler rating developed.

The overall *boiler efficiency* is the ratio of the heat absorbed to that supplied, i.e., output divided by input, or

$$e_b = \frac{W_s(h - h_f)}{W_f \times F} \times 100, \tag{81}$$

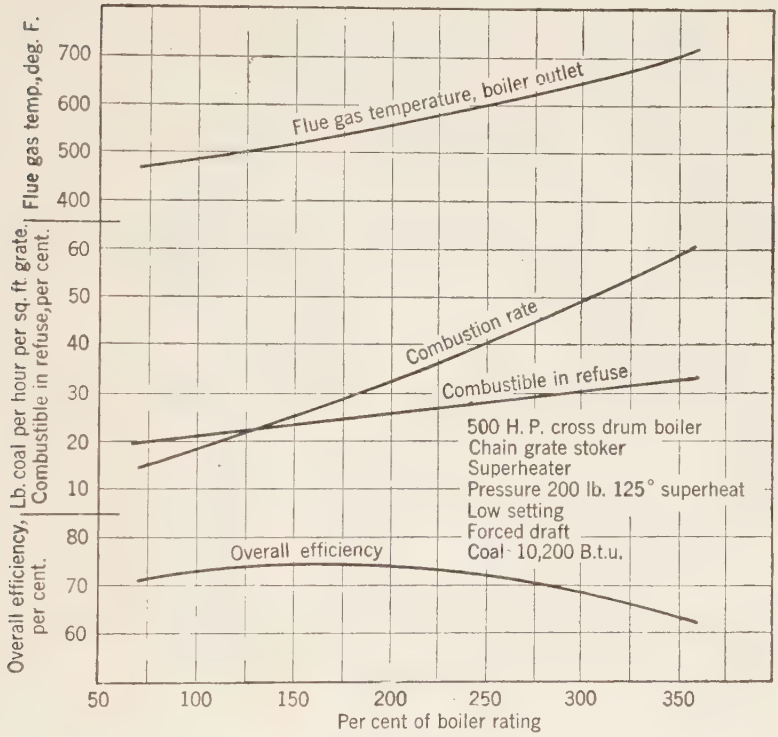

Flue gas temp., deg. F.
Lb. coal per hour per sq. ft. grate, Combustible in refuse, per cent.
Overall efficiency, per cent.
Flue gas temperature, boiler outlet
Combustion rate
Combustible in refuse
Overall efficiency
500 H. P. cross drum boiler
Chain grate stoker
Superheater
Pressure 200 lb. 125° superheat
Low setting
Forced draft
Coal 10,200 B.t.u.
Per cent of boiler rating

FIG. 44.—Typical boiler performance curves.

in which e_b = efficiency of steam-generating unit, per cent; includes boiler, superheater, furnace, water walls, grate, air heater, and economizer.

W_f = total weight of fuel fired per hour, lb.

F = high heat value of fuel as fired, B.t.u. per lb.

The rate of combustion may be expressed in pounds of fuel burned (*a*) per square foot of projected grate surface per hour, or (*b*) per cubic foot of furnace volume per hour. The boiler horsepower that can be developed is dependent on the rate of combustion, i.e., upon the kind of fuel, size and kind of grate, and quantity of air supplied. Rates of

combustion as high as 200 lb. of coal per sq. ft. of grate surface per hour have been obtained in locomotive boilers with a resultant high capacity, but a lowered efficiency. In modern steam power plants, the rate of combustion for a given boiler setting for maximum efficiency is determined experimentally. The efficiency increases with the capacity (boiler horsepower) and rate of combustion until a maximum is reached; beyond this point the efficiency decreases with increase of capacity; see Figs. 44 and 45. In Fig. 44 the efficiency peak occurs between 125

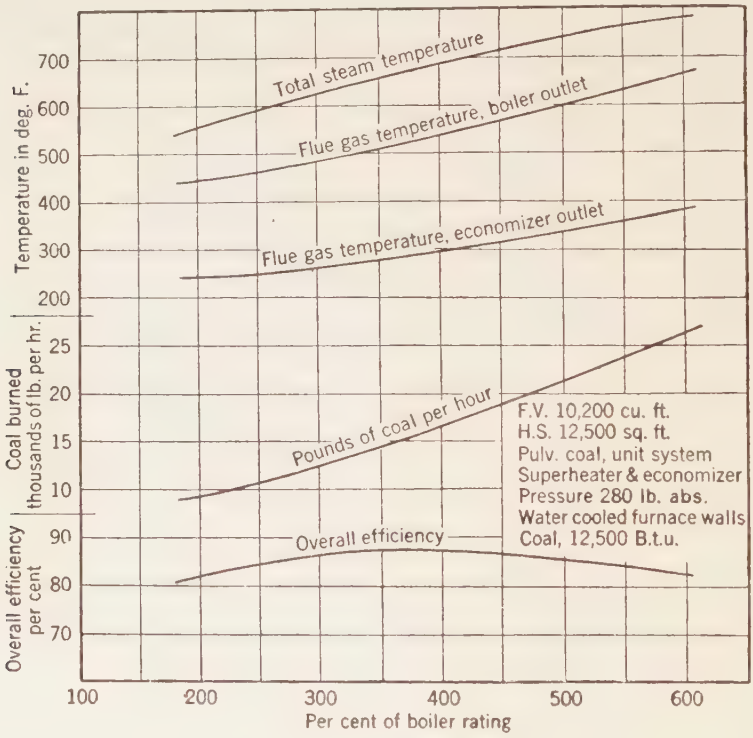


Fig. 45.—Performance of boiler No. 15; C. R. Huntley Station, Buffalo, New York.

and 225 per cent rating, beyond which the efficiency decreases because of the heat lost in increased flue-gas temperature and an increase in the combustible (unburned carbon) in the refuse. A boiler having a pulverized-coal furnace and an economizer showed an efficiency peak, Fig. 45, between 300 and 400 per cent of boiler rating. Comparison of the curves of the flue-gas temperature shows how the use of an economizer decreases the heat carried away by the flue gases.

Small hand-fired boiler equipment may operate with overall efficiencies varying from 40 to 65 per cent. Stoker-fired boilers may show

from 60 to 85 per cent efficiency, depending upon the size of the plant and the amount of heat-saving equipment in operation. Large pulverized-coal installations have shown boiler efficiencies of 80 to 90 per cent. If air preheaters or economizers are in use in such plants, overall efficiencies of 82 to 92 per cent may be obtained at times. Large oil-burning boiler plants show efficiencies between those of stoker-fired and pulverized-coal equipment. It should be noted that the average all-year boiler-operating efficiencies are well below the high efficiencies obtained for short periods of time during test conditions. In steam-boiler practice the heat not utilized per pound of fuel generally varies from 15 to 30 per cent of the calorific value of the fuel.

Example.—A boiler evaporated 40,000 lb. of water per hour from a feed-water temperature of 218 deg. F. to steam at 180 lb. per sq. in. (absolute pressure) and a quality of 97 per cent. Weight of coal fired per hour, 4500 lb., high heat value of the coal as fired 11,800 B.t.u. per lb. Required: the rate of heat absorption in kilo B.t.u. per hour, boiler horsepower developed, and the overall boiler efficiency.

Solution.—Substituting in (79),

$$Q = \frac{40{,}000\,[345.99 + (0.97 \times 850.3) - 186.04]}{1000},$$

$$Q = 39{,}389.64 \text{ kB. per hour.}$$

Substituting in (80),

$$\textit{Boiler hp.} = \frac{40{,}000\,[345.99 + (0.97 \times 850.3) - 186.04]}{970.2 \times 34.5},$$

$$\textit{Boiler hp.} = 1176.8.$$

Substituting in (81),

$$e_b = \frac{40{,}000\,[345.99 + (0.97 \times 850.3) - 186.04]}{4500 \times 11{,}800} \times 100,$$

$$e_b = 74.2 \text{ per cent.}$$

96. Boiler and Furnace Heat Balance.—The distribution of heat resulting from the combustion of fuel in a boiler furnace can best be studied by means of a *heat balance*, which constitutes a tabulation of the various heat losses and the heat absorbed by the boiler. The items considered may be listed as follows:

1. Heat absorbed by steam-generating unit.
2. Heat loss due to moisture in fuel.
3. Heat loss due to water from combustion of hydrogen.
4. Heat loss due to moisture in air supplied.
5. Heat loss due to dry chimney gases.

6. Heat loss due to incomplete combustion of carbon.
7. Heat loss due to unconsumed combustible in refuse.
8. Heat loss due to unconsumed hydrogen and hydrocarbons and to radiation and heat loss unaccounted for.

Calculation of the above items is generally made in terms of B.t.u. per pound of fuel and in percentage. Hence, the totals should equal the high heat value of the fuel as fired and 100 per cent, respectively.

1. *Heat absorbed by steam-generating unit.*—The heat absorbed by the boiler may be calculated by the following equation,

$$H_1 = \frac{W_s}{W_f}(h - h_f), \tag{82}$$

in which H_1 = B.t.u. absorbed by water and steam per pound of fuel.

2. *Heat loss due to moisture in fuel.*—The moisture in the fuel is evaporated and leaves the boiler setting in the form of superheated steam. Assuming that evaporation of the moisture occurs at 100 deg. F., because of the low vapor pressure exerted by the water in the flue gases,

$$H_2 = W_m[1104.2 + 0.455(t_g - 100) - (t_f - 32)],$$

or

$$H_2 = W_m(1090.7 + 0.455t_g - t_f); \tag{83}$$

in which H_2 = B.t.u. loss per pound of fuel.
W_m = weight of free moisture per pound of fuel.
t_g = temperature of the flue gases, deg. F.
t_f = temperature of the fuel entering furnace, deg. F.

3. *Heat loss due to water from combustion of hydrogen.* The hydrogen in the fuel burns to water, which leaves the boiler setting in the form of superheated steam.

$$H_3 = 9H_y(1090.7 + 0.455t_g - t_f), \tag{84}$$

in which H_3 = B.t.u. loss per pound of fuel.
H_y = weight of hydrogen per pound of fuel.

4. *Heat loss due to moisture in air supplied.*—This heat loss is small and may be computed thus:

$$H_4 = W_v(t_g - t_a) \times 0.47, \tag{85}$$

in which H_4 = B.t.u. loss per pound of fuel.
W_v = weight of water vapor supplied with air per pound of fuel.

W_v = relative humidity, expressed decimally, × weight of water vapor required to saturate one pound of dry air (from Table XXIV) at t_a × weight of dry air used per pound of fuel.

t_a = temperature of air entering furnace, deg. F.

5. *Heat loss due to dry chimney gases.*—The heat lost in the dry chimney gases is usually the largest loss and may be calculated by the following equation,

$$H_5 = W_{dg}c_p(t_g - t_a), \tag{86}$$

in which H_5 = B.t.u. loss per pound of fuel.

W_{dg} = weight of dry flue gas per pound of fuel.

c_p = mean specific heat of dry gas (about 0.24).

6. *Heat loss due to incomplete combustion of carbon.*—This heat loss is generally small and is caused by an insufficient air supply, some of the carbon in the fuel being burned to carbon monoxide.

$$H_6 = \frac{CO}{CO_2 + CO}(14{,}600 - 4440)C_1,$$

or

$$H_6 = \frac{10{,}160CO}{CO_2 + CO} \times C_1; \tag{87}$$

in which H_6 = B.t.u. loss per pound of fuel.

CO_2 and CO = percentages by volume as determined by the flue-gas analysis.

C_1 = weight of carbon actually burned per pound of fuel.

7. *Heat loss due to unconsumed combustible in the refuse.*—Some of the carbon in the fuel, either unburned or partially burned, may fall into the ashpit. This loss depends upon the type of grate, rate of combustion, and size and kind of coal.

$$H_7 = \frac{14{,}600W_rC_r}{W_f}, \tag{88}$$

in which H_7 = B.t.u. loss per pound of fuel.

W_r = weight of refuse, lb. per hour.

C_r = weight of carbon per pound of refuse.

8. *Heat loss due to unconsumed hydrogen and hydrocarbons, and to radiation and heat loss unaccounted for.*—These losses are determined by subtracting the heat absorbed by the boiler and heat losses 2 to 6 inclu-

sive from the high heat value per pound of fuel. Unaccounted-for heat losses are those which cannot be measured.

$$H_8 = F - (H_1 + H_2 + H_3 + H_4 + H_5 + H_6 + H_7). \qquad (89)$$

Example.—Tabulate the various heat losses and show method of calculation from the following data:

Heat absorbed by boiler, 74.8 per cent of the high heat value of the fuel.

Analysis of coal as fired, per cent by weight:

Carbon	62.0
Hydrogen	4.0
Oxygen	8.0
Sulphur	2.0
Nitrogen	1.0
Free moisture	8.0
Ash	15.0
Total	100.0

High heat value of coal as fired, 11,800 B.t.u. per lb.

Flue gas analysis, per cent by volume:

CO_2	13.0
CO	1.0
O_2	6.0
N_2 (by difference)	80.0

Temperature of air and fuel entering furnace, 74 deg. F.; temperature of flue gases, 452 deg. F.; temperature of steam in boiler, 360 deg. F.; relative humidity of air entering furnace, 70 per cent; coal fired per hour, 2000 lb.; refuse per hour, 420 lb.; and unburned carbon in refuse 18 per cent. Barometer, 29.92 in. of mercury.

Solution.—The actual weight of carbon appearing in the flue gas from equation 72 is

$$C_1 = \frac{(2000 \times 62) - (420 \times 18)}{2000 \times 100} = 0.5822 \text{ lb. per lb. of coal.}$$

Weight of dry chimney gases, equation 76,

$$W_{dg} = \left[\frac{(4 \times 13) + 6 + 700}{3\,(13 + 1)}\right] \times 0.5822 = 10.51 \text{ lb. per lb. of coal.}$$

Weight of dry air supplied, equation 77,

$$W_{aa} = 10.51 - 0.5822 + 8\left(0.04 - \frac{0.08}{8}\right) = 10.17 \text{ lb. per lb. of coal.}$$

HEAT LOSSES PER POUND OF COAL AS FIRED

Name of Heat Loss	Method of Calculation	B.t.u.	Per Cent
Heat absorbed by boiler	$0.748 \times 11{,}800$	8826	74.80
Moisture in fuel	$0.08\,[1090.7 + (0.455 \times 452) - 74]$	98	0.83
Hydrogen in fuel	$9 \times 0.04\,[1090.7 + (0.455 \times 452) - 74]$	440	3.73
Water vapor in air	$0.70 \times 0.01815 \times 10.17\,(452 - 74)\,0.47$	23	0.18
Dry chimney gases	$10.51 \times 0.24\,(452 - 74)$	954	8.08
Incomplete combustion	$[(10{,}160 \times 0.01) \div (0.13 + 0.01)]\,0.5822$	422	3.58
Combustible in refuse	$(14{,}600 \times 420 \times 0.18) \div 2000$	552	4.68
Radiation, etc.	By difference	485	4.12
	Total	11,800	100.00

The heat balance as ordinarily calculated gives only the distribution of the *actual* losses for boiler operation over a limited period of time. It does not represent average operating conditions, since the various *standby* losses are not considered. These include: (1) heat lost in shutting down boilers; (2) fuel required to start up cold boilers; (3) fuel burned in banking fires; and (4) heat discharged to waste in "blowing off" and for cleaning boiler surfaces. Some of the actual heat losses may be considerably reduced or entirely eliminated; other losses are *inherent* and cannot be prevented. Inherent losses are due to heating the flue gases, moisture in fuel, moisture in air supplied for combustion, and the moisture from the combustion of hydrogen from boiler-room temperature to steam temperature.

97. Boiler Accessories.—All devices which are directly connected with a boiler or which facilitate its operation and maintenance are called *boiler accessories*. The accessories usually provided are: steam-pressure gage, water column, feedwater regulator, safety valve, fusible plug, blow-offs, mechanical tube cleaner, and soot blower.

The steam pressure in a boiler is usually indicated by a *pressure gage*. Pressure gages in general use are of two classes, known as the Bourdon and diaphragm gages. In the Bourdon gage, Fig. 46, the pressure is exerted on the interior of a metal tube, oval in cross-section, bent to fit into a circular case. The application of pressure tends to make the cross-section round and thus to unroll or straighten the tube; this motion, communicated by means of links, levers, a gear sector, and a pinion, rotates a hand above a calibrated dial, and the pressure, in pounds per square inch above that of the atmosphere, may be read. In the diaphragm gage the pressure is resisted by a corrugated disk or diaphragm. The motion is transmitted to the hand in the same man-

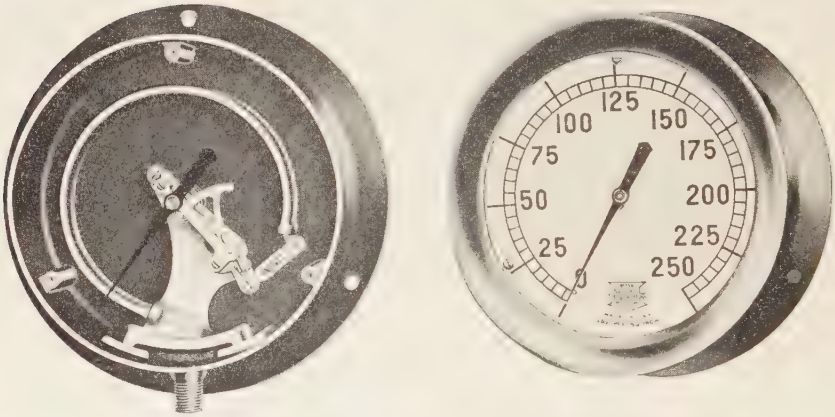


FIG. 46.—Two views of a typical Bourdon tube pressure gage.

ner as in the Bourdon gage. These gages (Bourdon and diaphragm) are affected by heat and hence should have a bent tube or equivalent, containing water, between the gage and the steam space. These protecting devices are called siphons.

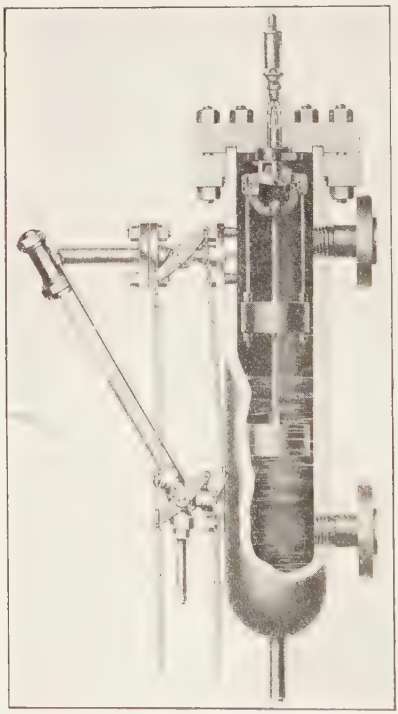


FIG. 47.—Yarway floatless "Hi-lo" alarm water column and "Se-sure" inclined water gage.

The *water column* and attached *gage glass*, Fig. 47, are connected to the front of the boiler drum, so that the water level may be visible from the boiler-room floor. The top of the water column is connected with the steam space and the bottom of the water column to the water space in the boiler as shown. Mud and sediment are removed by blow-off cocks attached to the lower end of the water column and gage glass. Gage cocks (not shown in Fig. 47) are used as a check on the indicated water level. The water column shown is provided with high- and low-water safety alarms. The action of the water column depends upon the displacement of solid weights suspended from the opposite ends of two levers, the whole assembly being in static equilibrium at normal boiler water levels with the alarm valve seated. Whenever

high- or low-water conditions prevail, the equilibrium of the weights is changed; either condition opens the valve located at the top of the column, and the released steam blows the whistle or operates an electric alarm.

The water level in some horizontal water-tube boilers is maintained at a point where the drum is about half full of water. The water column is usually connected so that the gage glass is half filled at the desired water level. The gage glass, Fig. 47, is inclined at an angle of 30 deg. from the vertical; thus, the water level can be seen readily from any position on the boiler-room floor.

The purpose of a *feedwater regulator* is to supply automatically a boiler with the same amount of water as is being converted into steam, thereby maintaining a constant water level.

The following types of feedwater regulators are in common use: float-lever, thermo-expansion, and thermo-pressure. They are generally used in connection with suitable pump governors. In the float-lever type the actuating member is a hollow float which rises and falls with the water level. When the water level falls, Fig. 48, the movement of the float opens the feedwater control valve through levers, and more feedwater enters the boiler. The float is open at the top and is kept filled with water by condensate which is continually forming in the steam space above and falling into the small pan. The weighted balance gives the water-filled float its buoyancy and maintains the center line of the float coincident with the water level; the mean water level in both the regulator and boiler corresponds. This type of regulator responds quickly to fluctuations of water level and is built for boiler pressures up to 700 lb. per sq. in.

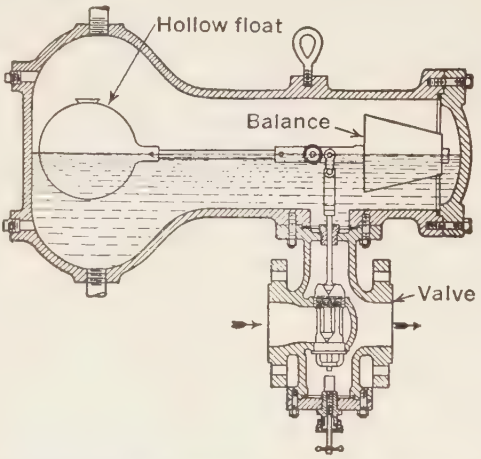


FIG. 48.—Stets float-type feedwater regulator.

In the thermo-expansion type of feedwater regulator, the expansions and contractions of an inclined metal tube are produced by the fall or rise in the water level. As Fig. 49 shows, the water level in the inclined expansion tube is the same as the water level in the boiler. With the boiler water at normal level, half of the expansion tube is filled with steam and half with water. When the steam output of the boiler

increases, the water in the boiler and tube begins to fall; more steam is now contained in the tube. The tube then expands and opens the control valve through a system of levers, thus increasing the rate of

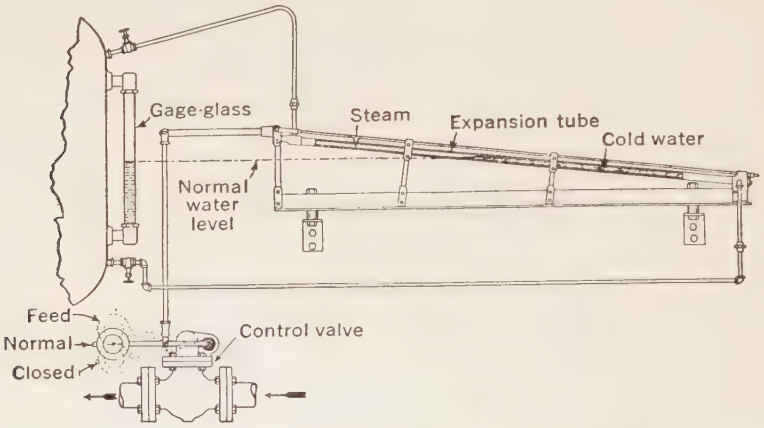


Fig. 49.—Copes thermo-expansion feedwater regulator.

feedwater flow. When the water level is high, the expansion tube contracts, thus closing the control valve.

Safety valves are used to protect boilers against excessive steam pressures, by opening automatically at a desired pressure and allowing the steam to escape. Figure 50 shows a section through a Pop Safety Valve (the only type sanctioned by the A.S.M.E. Boiler Code) in which the boiler pressure is resisted by a spring. The spring tension may be adjusted at the top and a lever is provided for manually testing the valve. The number and size of safety valves for a given boiler are usually specified by city or state legislation, or by insurance companies.

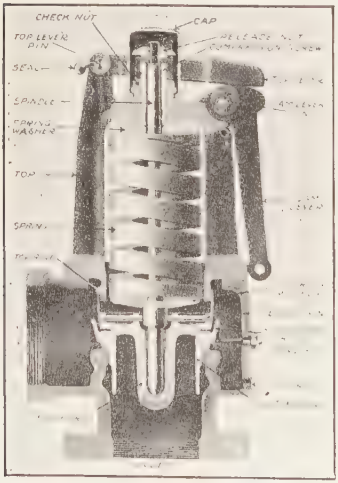


Fig. 50. Pop safety valve.

Fusible plugs, Fig. 51, are recommended to protect boilers operating at less than 225 lb. per sq. in. pressure. They consist of steel or bronze bushings, filled with tin alloy which melts at or about 450 deg. F., and are inserted in the boiler plate at the lowest permissible water-level location

as listed by the A.S.M.E. Boiler Code. The melting point of the tin is above the temperature of the steam and below the temperature of the hot gases. The small end of the plug, Fig. 51, is exposed to the products of combustion; when the water level is low enough to uncover the plug, the alloy melts and steam escapes. In this way, probable excessive boiler pressure is prevented and the escaping steam attracts the attention of the attendant, so that precautions may be taken to prevent overheating of the boiler metal.

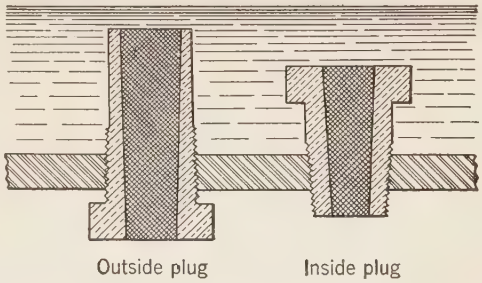


FIG. 51.—Typical fusible plugs which conform to A.S.M.E. boiler code.

Blow-off connections, as shown in the boiler illustrations of Figs. 27 to 42, made at the lowest part of the boiler and sometimes also to the steam drum, are used for the removal of water to carry off mud, sediment, scale, scum, etc. Impurities from the large quantities of water evaporated are constantly being precipitated owing to the high temperatures in boilers. A continuous (bottom) blow-off using a small pipe is sometimes used to remove the impurities as they are precipitated; however, when this is done the larger blow-off valves must be opened occasionally for a more thorough removal of mud, etc.

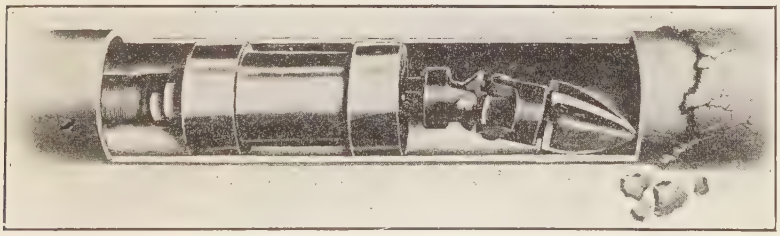


FIG. 52.—Liberty hammer-type tube cleaner.

Feedwater impurities combine mechanically and chemically, during boiler operation, to form a layer of scale on the heating surfaces. The methods used to decrease the formation of scale by purification of the feedwater are discussed in Chapter IX. The heat loss due to boiler scale varies with the temperature and rate of gas and water flow. Experiments [4] have shown a heat conductivity loss of about 12 per cent

[4] "Heat Transmission Through Boiler Tubes," by H. O. Croft; University of Illinois Engineering Experiment Station, Bulletin 168, 1927.

for $\frac{1}{16}$-in. thickness of scale, for boiler tubes (lower rows) near the furnace. Scale may be the cause of localized overheating, resulting

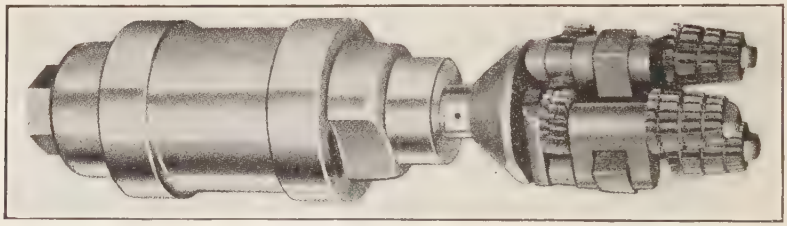


FIG. 53.—Lagonda revolving-cutter type tube cleaner.

in serious burning, blistering, and damaging of tubes and plates. All boilers require the use of a mechanical tube cleaner at definite

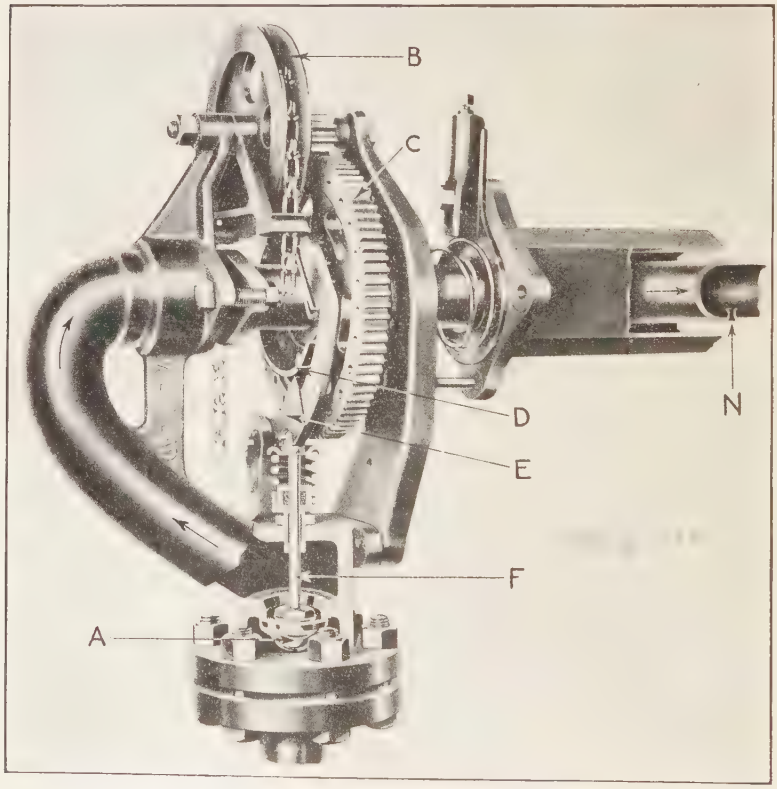


FIG. 54.—Diamond mechanical soot blower.

intervals in order to remove effectively the scale formed on the tube surfaces. *Mechanical tube cleaners* may be divided into two classes:

(1) *hammer* type, Fig. 52, which loosens the scale by rapid blows, is applicable to water- and fire-tube boilers; (2) *revolving cutter type*, Fig. 53, removes the scale by rotating cutting tools and is used in water tubes only. The most popular type of tube cleaner today is the air-driven cleaner (60 to 90 lb. pressure), because it is possible to obtain greater power and speed with air than by steam or water under pressure.

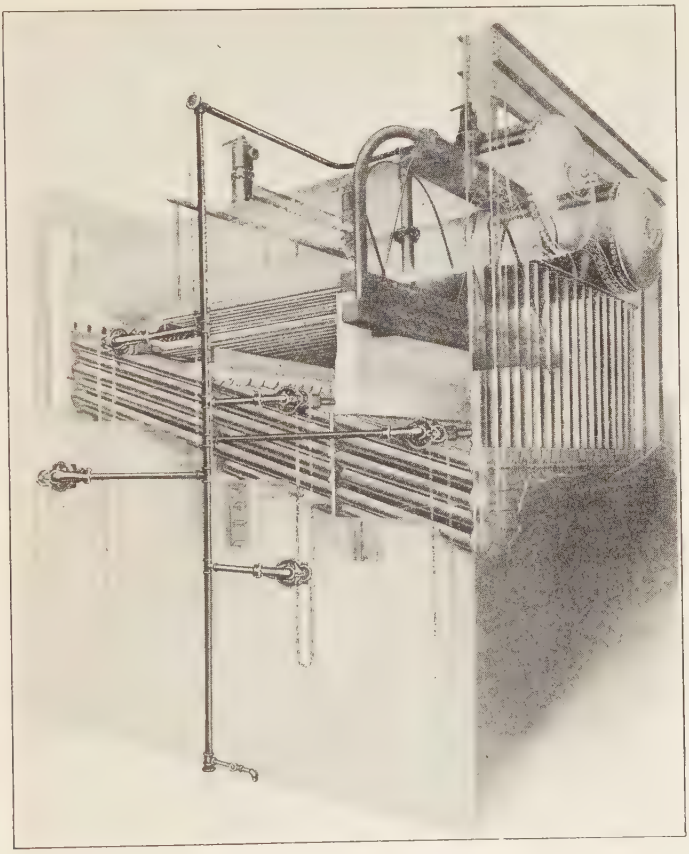


Fig. 55.—Typical installation of mechanical soot blowers.

Soot is an excellent heat-insulating material, and hence its removal from boiler heating surfaces is desirable. Soot is broadly understood to include all the solid matter leaving the fuel bed with the gaseous products of combustion. Boiler surfaces exposed to the flue gases are usually freed from ash and soot by hand-operated steam lances, mechanical soot blowers, brushes, or similar devices. *Mechanical soot blowers*, Fig. 54, are permanently installed in boilers, as shown in Fig. 55, and so

located that all parts of the heating surface subjected to soot deposit may be cleaned with jets of steam, air, or a mixture of air and steam. With the soot blower shown by Fig. 54, pulling the operating chain automatically opens valve A and revolves the blower element N. Sheave B rotates gear C, and valve A is opened through cam D and trigger E; steam then flows through the blower head to the nozzles N, as indicated by the arrows. The soot blower is designed to rotate in one direction, sweeping over the blowing arc once every revolution; and when the blowing arc is completed, the steam flow is shut off by cam D. The operator may revolve the nozzle through any number of blowing arcs required to effect the cleaning.

PROBLEMS

1. A boiler generated dry steam at 190 lb. per sq. in. gage. The feedwater was supplied at a temperature of 78 deg. F., and the water evaporated per hour was 16,000 lb. Barometer 29.60 in. of mercury. Required the heat absorbed by the water and steam in kilo B.t.u. per hour, and the boiler horsepower developed.

2. A boiler was operated at 290 per cent of its rated capacity; how many pounds of water were evaporated from and at 212 deg. F. per square foot of heating surface per hour?

3. A 600-hp. unaflow steam engine used 13.5 lb. of steam of 98 per cent quality per horsepower per hour. The pressure at the boiler was 180 lb. per sq. in. abs., and the feedwater temperature was 182 deg. F. (*a*) What boiler horsepower was necessary to supply steam to this engine? (*b*) Find the ratio of steam engine horsepower to boiler horsepower developed.

4. Two boilers were connected to the same steam header. One boiler supplied 7000 lb. of steam per hour containing 2.3 per cent moisture; the other, 6000 lb. per hour which was superheated 120 deg. F. The absolute pressure in the header was 170 lb. per sq. in. Find the quality or condition of the steam in the header.

5. A boiler with 1000 sq. ft. of heating surface generated 8000 lb. of steam per hour at 120 lb. per sq. in. abs. pressure and 100 deg. F. superheat, from feedwater at 160 deg. F. Calculate the horsepower developed and the percentage of rating at which the boiler was operated.

6. Seven pounds of dry saturated steam at 140 lb. per sq. in. abs. were generated per pound of coal from feedwater at 120 deg. F. Coal composition C, 60; H, 4; O, 8; N, 1; S, 3; M, 14; A, 10 per cent. Calculate the heat value of the fuel. What was the overall boiler efficiency?

7. Determine the overall efficiency of a boiler that developed 250 per cent of its rating when coal of 13,000 B.t.u. per lb. was burned at the rate of 35 lb. per sq. ft. of grate surface per hour. Assume one square foot of grate surface equivalent to 4.5 rated boiler horsepower.

8. Find the ratio of grate surface to rated boiler horsepower for a boiler which developed 150 per cent rating and 70 per cent efficiency with a combustion rate of 25 lb. per sq. ft. of grate surface per hour. Coal had 13,200 B.t.u. per lb.

9. By test a steam generator showed an actual evaporation of 13.5 lb. of water per lb. of fuel oil. The feedwater temperature was 208 deg. F.; the steam pressure 250 lb. per sq. in. abs.; and the temperature of the steam 500 deg. F. Determine the overall efficiency, if the calorific value of the fuel oil was 18,500 B.t.u. per lb.

10. Fuels of the following heat values per pound were available for use: oil, 18,200 B.t.u.; coal, 12,800 B.t.u.; and lignite, 7200 B.t.u. The cost of fuel oil was $1.40 per barrel of 42 gallons, specific gravity, 0.87; of coal, $3.50 per ton; and of lignite, $1.95 per ton. The boiler efficiencies were 80 per cent for oil; 70 per cent for coal; and 60 per cent for lignite. Calculate the cost, with each fuel, to evaporate 1000 lb. of water from and at 212 deg. F.

11. The equivalent evaporation of a boiler having a heating surface of 10,000 sq. ft. was 65,000 lb. of water per hour from and at 212 deg. F. The boiler showed an efficiency of 85 per cent using pulverized coal as the fuel. Find the number of B.t.u. liberated by the burning of the fuel per cubic foot of furnace volume per hour, if the furnace volume was 8000 cu. ft.

12. A boiler operating at 200 per cent of its rated capacity showed an efficiency of 82 per cent when using coal with a heat value of 14,000 B.t.u. per lb. If the ratio of heating surface to grate surface was 50 to 1, find the rate of combustion in pounds of coal per square foot of grate surface per hour.

13. An oil-fired boiler gave an actual evaporation of 12 lb. of water per lb. of oil with an overall efficiency of 78 per cent. Feedwater temperature 162 deg. F., steam pressure 200 lb. abs., and quality of steam leaving the boiler 98 per cent. (*a*) Required the heating value of the fuel. (*b*) If the fuel oil cost five cents per gallon (7.5 lb. per gal.), determine the fuel cost for evaporating 1000 lb. of water from and at 212 deg. F.

14. The steam used by a turbine was 15 lb. per kilowatt-hour, at a pressure of 400 lb. per sq. in. abs. and a temperature of 550 deg. F. The overall boiler efficiency was 82 per cent with feedwater at 200 deg. F. (*a*) How many pounds of 13,000 B.t.u. coal were required per kilowatt-hour? (*b*) If coal cost $3.40 per ton what was the fuel cost per kilowatt-hour?

15. The overall efficiency of a boiler was 72 per cent using coal with a heat value of 12,600 B.t.u. per lb. and costing $3.20 per ton. Steam was generated at a pressure of 150 lb. per sq. in. abs. and 142 deg. of superheat. (*a*) If the feedwater temperature was 60 deg. F. find the fuel cost per 1000 lb. of steam generated. (*b*) What would be the cost if the feedwater temperature were 200 deg. F.? (*c*) If it were 285 deg. F.? (*d*) What would be the cost if the conditions were as in *a* except that the efficiency equaled 82 per cent?

16. A boiler required 2350 lb. of coal per hour; coal heat value 14,200 B.t.u. per lb. Temperature of feedwater, 192 deg. F.; steam pressure in boiler, 200 lb. per sq. in. gage. A throttling calorimeter showed a temperature of 252 deg. F. at an atmospheric pressure of 29.92 in. mercury. The overall boiler efficiency was 65 per cent. (*a*) Find the kilo B.t.u. absorbed per hour. (*b*) How many pounds of steam were generated per hour? (*c*) What boiler horsepower was being developed?

17. During a 24-hour period 240,000 lb. of water were fed to a boiler and 32,000 lb. of coal were fired. The temperature of the feedwater averaged 140 deg. F., steam pressure 160 lb. abs., steam quality 98.5 per cent, heat value of coal as received 11,700 B.t.u. per lb., moisture in the coal 10 per cent. (*a*) Determine the equivalent evaporation per pound of dry coal. (*b*) Find the horsepower developed. (*c*) What was the overall boiler efficiency? (*d*) How much could be paid for a feedwater heater (based on one year's savings) that would increase the feedwater temperature to 210 deg. F.? Interest on investment at 6 per cent and depreciation at 5 per cent. Assume coal cost $3.75 per ton and that the plant would operate 10 hours per day and 300 days per year.

18. A boiler evaporated 20,000 lb. of water per hour from a feedwater temperature of 209 deg. F. to dry steam at 350 lb. per sq. in. abs. pressure. If the boiler

developed 200 per cent of its rating when evaporating this amount of water, find the square feet of heating surface.

19. The following data were obtained from a boiler test: water fed to boiler 24,700 lb. per hour; feedwater temperature 62 deg. F.; steam pressure 180 lb. per sq. in. gage; quality of steam 97 per cent; and barometer 29.21 in. of mercury. (*a*) Find the kilo B.t.u. absorbed per hour. (*b*) Determine the number of square feet of heating surface if the boiler developed 150 per cent of its rating during this test.

20. The overall efficiency of a boiler was determined to be 72 per cent. Coal was burned at the rate of 500 lb. per hour; the moisture in the coal was 8 per cent; and the heating value per pound of dry coal was 12,600 B.t.u. The boiler pressure was 150 lb. per sq. in. abs ; the steam was discharged at a temperature of 400 deg. F., and the feedwater temperature was 202 deg. F. (*a*) How many pounds of steam were discharged per hour? (*b*) Find the kilo B.t.u. absorbed per hour. (*c*) Determine the equivalent evaporation, pounds per pound of dry coal. (*d*) If coal was purchased at $4.00 per ton, find the coal cost per thousand pounds of equivalent evaporation.

21. A boiler delivered 8800 lb. of steam per hour to the steam main at 180 lb. abs. pressure and 99 per cent quality with feedwater at 174 deg. F. (*a*) Determine the kilo B.t.u. absorbed per hour. (*b*) How many pounds of lignite were required per hour, if the overall boiler efficiency was 63 per cent and the calorific value of the lignite was 7000 B.t.u. per lb.?

22. If 22 per cent of the heat of combustion of natural gas containing 1000 B.t.u. per cu. ft. were lost, how many cubic feet of gas were required to evaporate 6760 lb. of water from an initial temperature of 142 deg. F. into dry steam under a pressure of 100 lb. per sq. in. abs.?

23. The temperature of a boiler room was 70 deg. F., the temperature of the flue gases 500 deg. F., and 2.5 times as much air was used as theoretically required. If the coal contained C, 74 per cent; H, 6 per cent; O, 4; N, 1; S, 2; M, 6; and A, 7 per cent; find the percentage of heat lost in the flue gases, based on the calculated heating value of the fuel.

24. The following data were obtained from a boiler test: Heating surface, 6000 sq. ft.; grate surface, 120 sq. ft.; furnace volume, 1200 cu. ft. Coal analysis, per cent by weight, as fired, C, 65; H, 5; N, 1; O, 8; S, 2; moisture, 8; ash, 11; and B.t.u. per lb., 11,980. Weights per hour: feedwater, 24,800 lb.; coal, 3150 lb.; dry refuse removed from ashpit, 550 lb.; combustible in dry refuse, 20 per cent. Pressures: steam pressure, 120 lb. gage; barometer, 29.84 in. Temperatures: feedwater, 150 deg. F.; steam, 350 deg. F.; flue gas, 450 deg. F.; boiler room, 80 deg. F.; relative humidity, 50 per cent. Flue-gas analysis, per cent by volume, CO_2, 14; CO, 0.8; O_2, 5.5; and N_2, 79.7. Find: (*a*) The kilo B.t.u. absorbed per hour. (*b*) The kilo B.t.u. absorbed per hour per square foot of heating surface. (*c*) The percentage of builders' rating developed. (*d*) The equivalent evaporation, pounds per pound of dry coal fired. (*e*) Pounds of combustible burned per hour per square foot of grate surface. (*f*) Combustion space per pound of combustible burned per hour. (*g*) The heat liberated per cubic foot of furnace volume. (*h*) The overall efficiency.

25. Using the data of problem 24, calculate and tabulate the complete boiler heat balance per pound of coal as fired. Show the method of calculation for each item.

CHAPTER VIII

STEAM-GENERATOR AUXILIARIES

98. Settings and Furnaces.—The enclosing and supporting structure for a boiler is called the *setting*. Boilers above 1000 sq. ft. of heating surface are usually supported by steelwork independent of the brickwork, Figs. 29, 30, and 31. That part of the setting where the fuel is burned is called the *furnace*. The generation of steam may be divided into two distinct operations: (1) the burning of the fuel, and (2) the heat absorption by the water and steam. Efficient utilization of the fuel is dependent to a large extent upon the correct design and construction of the furnace and setting as explained in Art. 73, Chapter VI.

The outer walls of a brick setting are usually built of hard red brick; the inner surface is faced with high-grade firebrick capable of withstanding high temperatures. Extremely high furnace temperatures are the rule today in the operation of large steam generators, and, consequently, firebrick must be of a quality that will stand up under severe conditions. However, to prolong the usefulness of the furnace lining, perforated brick blocks for air admission, radiant types of superheaters (Fig. 83), or water-circulating pipes (Fig. 68) have been installed in furnace walls. Insulating materials may be placed between the inner and outer brick walls to reduce the heat loss.

The boilers shown in Fig. 27 are said to be *internally fired* because the furnace is enclosed in the lower part of the boiler shell, and surrounded by an annular water leg. In an *externally fired* boiler the furnace is separate from the boiler, i.e., the fire is outside the shell, as in Fig. 29. Settings also include the foundation, ashpit, inspection doors, metal boiler fronts (opposite the headers), baffles, furnace arches, and heat-insulating covering for the tops.

The Dutch-oven furnace, Fig. 39, is located in an extension of the setting. Because the hot gases have a greater distance to travel before coming in contact with the cooler heating surfaces, more time is allowed for complete combustion. The roof of the furnace absorbs heat from the burning coal, becomes white hot, and radiates heat back upon the coal and in this way aids combustion.

99. Grates.—*A grate* is a sectional metallic structure designed to support the fuel in a furnace, and is so made that air for combustion can

pass through the openings to the fuel. Most grates are made up of sections of cast iron; cast iron is cheap and is not liable to be destroyed at high temperatures. The openings or air spaces depend upon the kind of fuel used, and the combined area of the openings varies from 20 to 40 per cent of the total grate area. The total grate area depends upon the heating value of the coal and the rate of combustion. Manufacturers usually provide about one square foot of grate surface for every 40 to 60 sq. ft. of water-heating surface.

Many types of grate bars have been designed according to various ideas as to the best methods of burning different fuels. Stationary grates for hand-fired furnaces are made of cast-iron sections about 3 ft. long and several inches wide, as shown in Fig. 56. The tupper and herringbone grate bars are stiffer and less likely to warp than the common bar, but they are not so convenient to use with coal that clinkers badly because of greater difficulty in slicing and cleaning of the fires. Lugs at each end of the bars rest upon bearer bars fastened to the walls of the setting. The length of the grate should not exceed 6 ft. for bituminous coal or 12 ft. for anthracite, otherwise the fireman cannot properly distribute the coal and break up the fire.

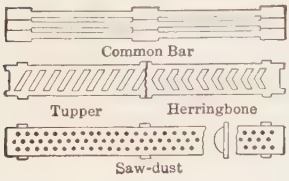


FIG. 56.—Types of grate bars for hand-fired furnaces.

For use in hand-fired plants, sectional *shaking* and *dumping grates* present a distinct advantage over stationary grates. The fireman can clean the fires without opening the furnace doors and without allowing the entrance of large quantities of excess air, which entails a waste of fuel and consequent low efficiency. For the boiler installation which is too small to warrant the use of a mechanical stoker, the shaking grate is often used. There are many types of shaking and dumping grates; in each, the grates are built up of a number of movable sections which are operated from the front of the furnace by one or more hand levers, as illustrated in Fig. 57. This design has been made as simple as possible by the elimination of small parts and complicated fastenings. The grate bars are of one-piece construction and are made with a lug at the lower end which is fastened to the shaking bar. The action of the grate is apparent from the illustration.

100. Mechanical Stokers.—A *mechanical stoker* is a power-operated fuel-feeding mechanism and grate. It is used for supplying solid fuel to a furnace, admitting air to the fuel for the purpose of combustion and providing a means for the removal of refuse. To accomplish this result several distinct types of stokers have been developed to suit the conditions of various installations, such as size of steam generator

served, fluctuations in load, draft available, quality of fuel used, and space limitations. The first cost and the cost of operation and repairs of a mechanical stoker are considerably greater than for hand firing, but less labor and attendance are required for operation. A cheaper grade of fuel can be used, a higher efficiency attained, greater flexibility of operation assured, better furnace conditions maintained, and less smoke formed than is usual with hand firing. Mechanical stokers are used in most power plants with units that generate more than 4000 lb. of steam per hour. Furnaces should have a volume of 20 (underfeed) to 30 (chain grate) cu. ft. per lb. of fuel burned per minute. Where coal, coke, or lignite is used as fuel, the types of completely automatic

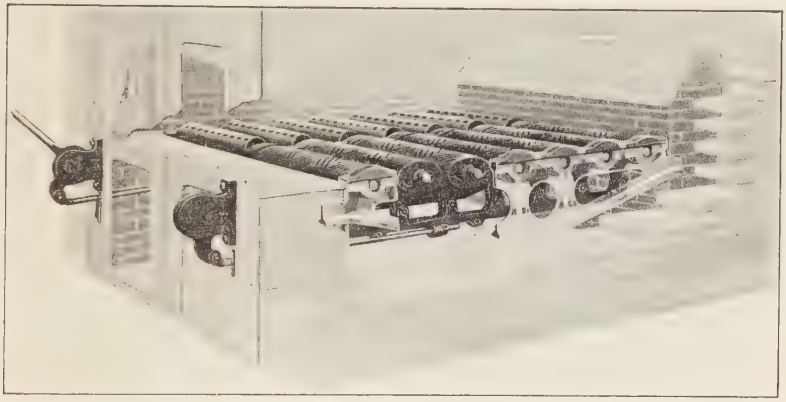


Fig. 57.—Thomas shaking grate.

mechanical stokers may be classified as chain-grate, overfeed, and underfeed.

101. Chain-grate Stokers.— In its elementary form, the chain-grate stoker consists of an endless grate composed of sections of bars passing over rotating sprockets or cylinders. The coal is fed from a hopper to the grate, Fig. 58, is ignited as it passes under an ignition arch, and is carried with the grate toward the rear of the furnace as the combustion progresses, gradually burning to ash. The ash and refuse are automatically dumped as the grate bars turn over the rear end of the stoker. Variation of the air supply and changing the speed of the grate provide definite means for controlling the progressive burning of each constituent of the fuel.

Stokers of this type have been widely used for burning bituminous coals which are of the non-coking type and are high in volatile matter and ash. In recent practice they have also been used to advantage with anthracite screenings, coke breeze, and lignite. Since the tars

of a coking coal fuse at a temperature lower than that of volatilization, such coals have a tendency to clog the air passages of the grate. For natural-draft chain grates using coals of this kind, a coking plate is provided upon which the coal drops from the hopper. There it remains until the volatile matter has been driven off, after which the coked coal is delivered to the grate. Forced-draft chain grates burn coking coals satisfactorily without agitation because the air supply can be regulated so as to consume the volatile matter where the fuel enters the furnace.

Chain-grate stokers are used with either natural or forced draft and are sometimes provided with draft compartments. These have

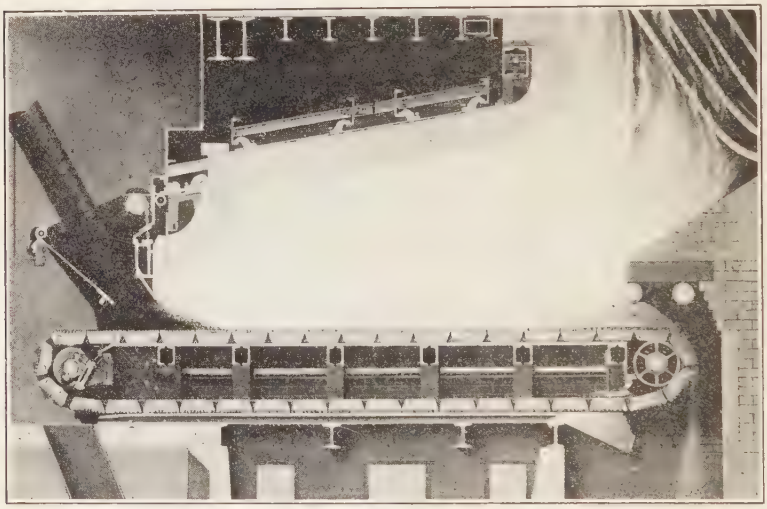


FIG. 58.—Side view of Green natural-draft chain-grate stoker under actual operating conditions.

individual damper control under the grate to give zone control of combustion, as shown in Fig. 59. At light loads this is an advantage, since it is possible to reduce the area on which the fuel is burned, thus giving a higher rate of combustion and usually higher efficiency. Chain-grate stokers are provided with an ignition arch to reflect the heat from the fire, down upon the fresh coal at the front of the grate. Furnace arches aid in maintaining the temperature for complete combustion before the gases reach the heating surface of the boiler. Incomplete combustion may occur with a chain grate when the fire is excessively forced. Extra air is likely to leak in if the fuel bed becomes too thin, thus causing a drop of temperature in the furnace, and poor combustion results.

Any leakage of air around the back end of the grate is a source of

loss, unless provision is made for heating this excess air and directing it into the path of the combustible gases. This leakage may be minimized by the use of a closed ashpit with a baffle placed just beyond the rear end of the stoker, a dump plate, or a water back placed on an overhung bridge wall; see Fig. 58. The water back, by presenting a watercooled surface to which clinkers will not adhere, retains the incandescent fuel on the grate until it is more thoroughly burned and eliminates fusing of the bridge wall. Modern chain-grate stoker installations are being provided with a rear arch, Fig. 36, in addition to the front arch; the rear arch deflects the excess air from the rear to the fuel bed

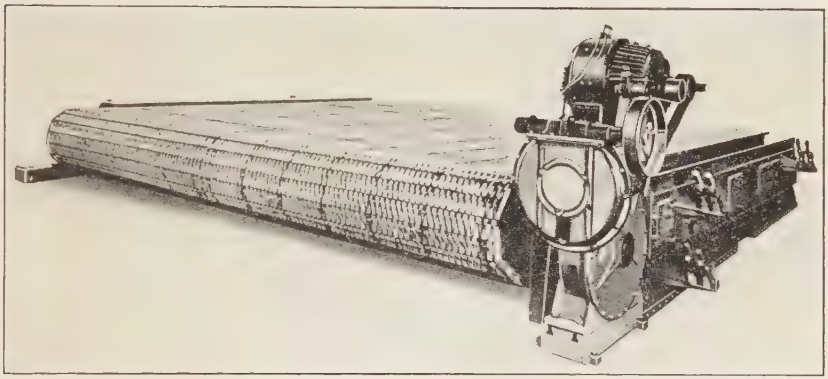


Fig. 59.—General view of Green forced-draft chain-grate stoker.

and into the combustion space. This deflection also prevents burning of the grate links.

At the rear of the coal hopper is a cast-iron plate lined with firebrick that may be raised or lowered, thus regulating the depth of the fuel bed. With the forced-draft chain-grate stoker the rate of combustion varies from 40 to 60 lb. per sq. ft. of grate surface, per hour, with probably 45 lb. as the average. Using natural draft, the average combustion rate is about 30 lb. per sq. ft. of rate surface per hour.

102. Overfeed Stokers.—Stokers of the overfeed type may be divided into two classes, the frontfeed, and the sidefeed (opposed-stoker). *Frontfeed stokers* receive coal from a hopper extending across the entire front of the furnace. In Fig. 60, coal is fed to the hopper by shovel or overhead conveyor and forced upon the arch jams by reciprocating pusher plates (mechanically operated). The fuel is exposed to the radiant heat of the furnace and ignition arch; some air for combustion enters from the space above the ignition arch. When the coal reaches the fuel bed, ignition has taken place and part of the volatile matter has been liberated. Air is supplied through space *E* and by

openings in and between the grate bars from the ashpit; additional air may be supplied by opening door *A*. Automatic rocking grate bars provide agitation of the fuel; the inclination of the grates is about 20 deg., which is sufficient, with the aid of the grate bar movement, to cause the fuel to descend. At the same time the working of the grate breaks up the coked fuel, providing air spaces through the fuel bed and sifting the ash to the space below. The coarser ash, clinker, and some unburned fuel reach the dump plate, at the rear of the furnace. This stoker feeds the coal mechanically and stokes it mechanically, but the

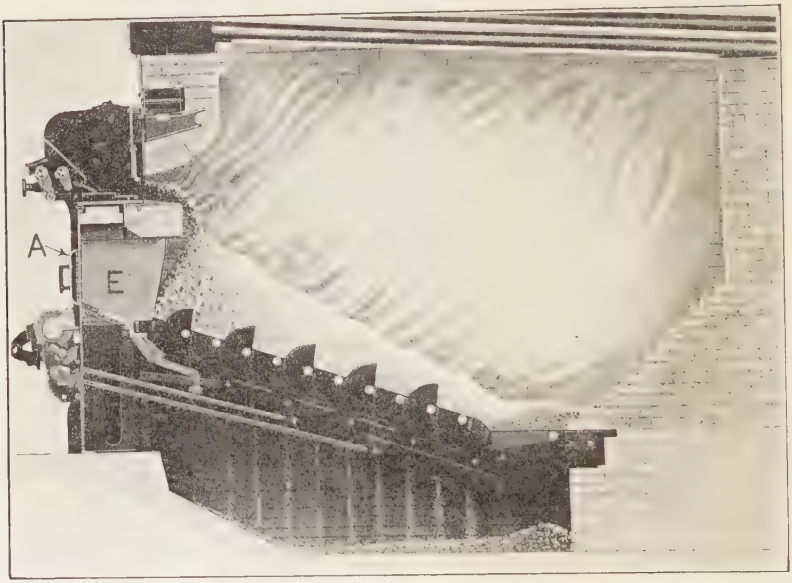


Fig. 60.—Huber overfeed stoker installation.

dump plate must be actuated manually for the removal of accumulated ash and refuse. The grate bars may be operated by a hand lever inserted in a bracket at the front of the setting. The pusher plates may also be actuated by hand in case of breakdown of the mechanical drive.

Fuel feed control is regulated by adjusting the stroke of the pusher plates for the grade of coal burned and by the motor speed for the load. The stoking control is such that the movement of the bars is comparatively fast so that they are not exposed in the hot fuel bed for more than a few seconds. The period between stoking is long enough to burn the coal added to the furnace during that time. The front of the furnace is made tight, so that forced draft may be used if desired. These stokers are adapted to low-volatile coking bituminous coals, such as are found in the eastern states, which require agitation of the fuel bed and

are easily ignited. When used with natural or induced draft, these stokers are not suitable to conditions where the load fluctuates rapidly. The maximum rate of combustion is about 40 lb. of coal per sq. ft. of grate surface per hour with forced draft, and about 20 lb. with natural draft.

Sidefeed stokers differ little in principle of operation from the frontfeed type. The Murphy stoker, Fig. 61, which is representative of this type, has coal hoppers *H* placed along the full length of each side of the furnace. This stoker uses two inclined grate surfaces so arranged

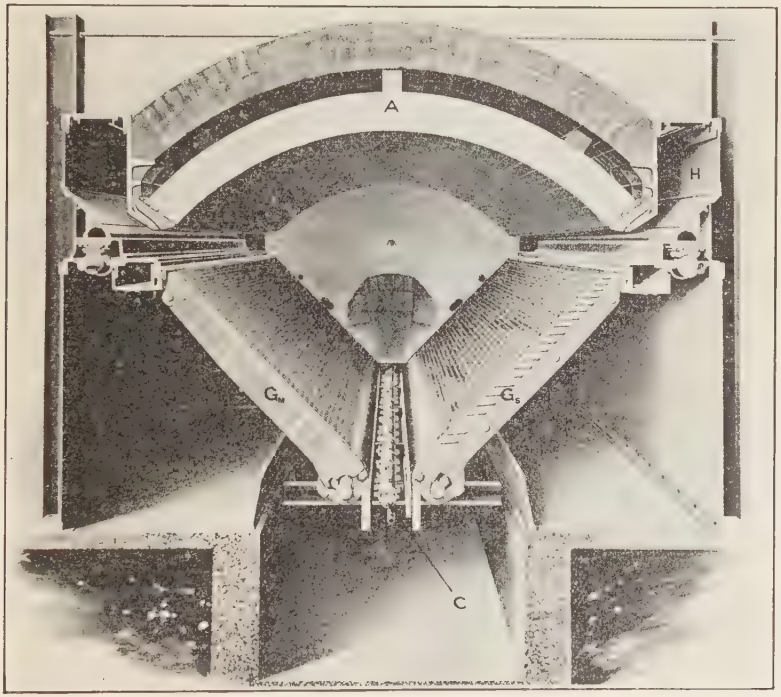


Fig. 61.—Murphy overfeed stoker, view from rear of furnace.

as to form a V-shaped inclined fuel bed, about 4 in. in maximum depth, which is agitated by alternate movable grate bars G_m, the bars being parallel with the travel of the fuel bed. The alternate stationary grate bars G_s are ribbed to prevent the entrance of excessive air from below and, further, to reduce sifting of fine coal to the space below. The furnace is always of the extended or Dutch-oven type with an air-cooled arch *A* covering the entire space above the grates. The process of combustion is essentially the same as that in frontfeed stokers, but in this case clinker grinders *C* are usually employed to remove the ash

automatically. The air used for cooling the arch is supplied (as preheated air) to the green coal through ducts in the arch plate. On account of the bottling effect of the furnace, the temperature runs higher than with the frontfeed type, making ignition more rapid but limiting the kinds of fuel used to those containing ash with a high-temperature fusing point. Rates of combustion are practically the same for sidefeed as for the frontfeed stokers. The Murphy stoker uses natural or induced draft and is particularly suited for small or medium-sized boilers carrying a steady load.

103. Underfeed Stokers.—Underfeed stokers are designed on the principle that the hottest part of the fuel bed should be located where its

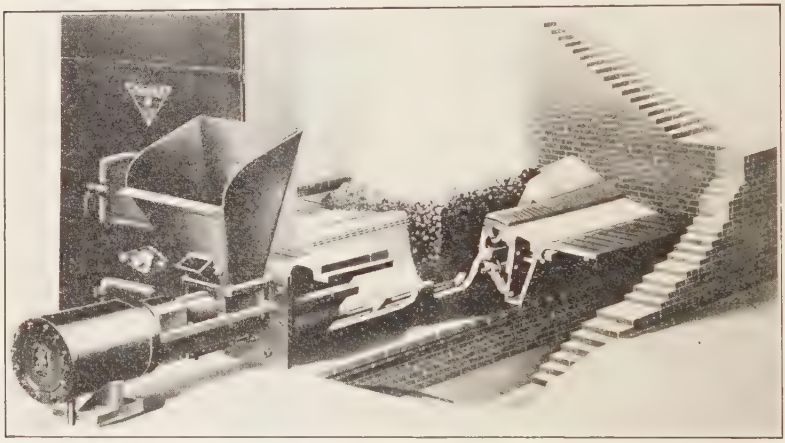


FIG. 62.—Jones side-dump underfeed stoker, single retort.

radiant heat will impinge directly upon the heating surfaces of the boiler. Coal is pushed from the hopper into a retort by means of a reciprocating plunger. The retort is similar in shape to a trough, see Fig. 62, with the green coal forced in along the bottom. The fuel is gradually pushed up to the surface during the process of coking and ignition, and overflows at the sides or end to the ash dump plate or clinker grinders, as provided.

As the fuel bed carried on underfeed stokers is usually 24 in. or more in thickness, forced draft is essential for their operation, the air pressure below the fire being as high as 5 to 7 in. of water. With underfeed stoker operation, there should always be a slight suction in the furnace. Air is forced into the fuel bed through tuyères in the side of the retort. In passing through the fuel bed the air becomes heated before supplying oxygen for the combustion, which is most intense at the upper surface of the fuel bed. The volatile gases driven off during the ascent of the

fuel in the retort pass through this incandescent bed at the surface and because of the forced draft are thoroughly mixed with air so that combustion of the gases is complete at a short distance above the fuel bed. Furnaces for underfeed stokers are not so large as others of equal coal-burning capacity and they require no special arches to secure ignition and proper mixture of furnace gases.

There are two general classes of underfeed stokers, the *single retort* and the *multiple retort*. Figure 62 shows a longitudinal section of a Jones side-dump underfeed stoker, one of the earliest and simplest forms of the single-retort type. Coal is fed from the hopper in front of a piston (shown in Fig. 64) and is moved forward and upward into the

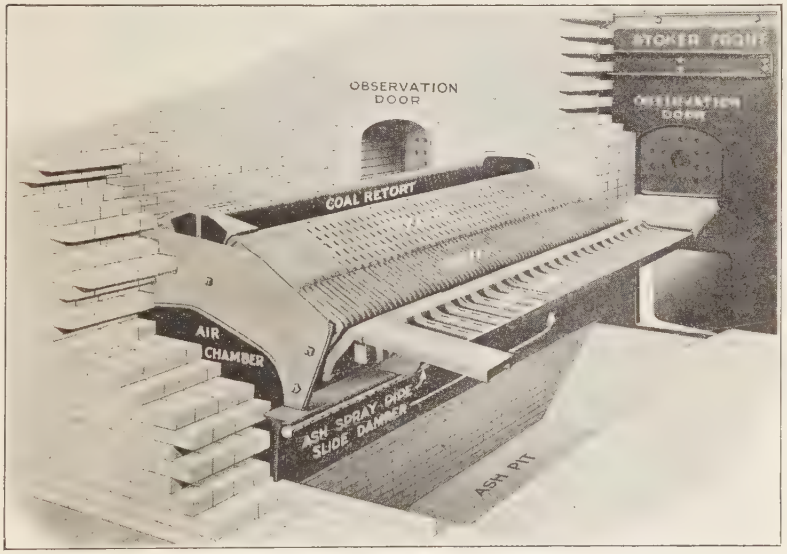


Fig. 63.—Furnace view of Detroit underfeed stoker.

retort by the reciprocating action of the steam-actuated plunger and the wedge-shaped pushers. In this way, coal is always introduced under the fire; air for combustion is forced into the sealed ashpit by a blower and passes into the furnace through tuyère openings in the hollow blast tiles located at a point above the fresh fuel in the retort, but below the fire. Thus, after the liberation of the volatile gases, the coke (fixed carbon) is burned at the top of the fire. The ash forms on top of the fire, is forced to side dump plates, and is dumped at intervals by a hand lever at the front of the furnace; the ash drops into a shallow ashpit from which it can be removed easily. One or more stokers (retorts) of this type may be installed in a furnace, depending upon the capacity of the boiler and width of the furnace. The supply of fuel and

air may be controlled automatically by the steam pressure in the boiler; the result is that the stoker, if correctly installed and operated, effects complete and smokeless combustion. The Jones stokers are especially suitable for use in smaller installations and have a fuel-burning capacity of 1200 to 1500 lb. of coal per hour per retort.

A furnace view of the Detroit single-retort stoker is shown by Fig. 63. Coal is forced into the retort by a plunger; from there it overflows onto the tuyères, grates, and ash dump. In this way the fuel is first coked and then completely burned, the refuse passing to the ash dump sections. A sealed air chamber extends beneath the entire length of the fuel bed, from which air is supplied to every part of the fuel

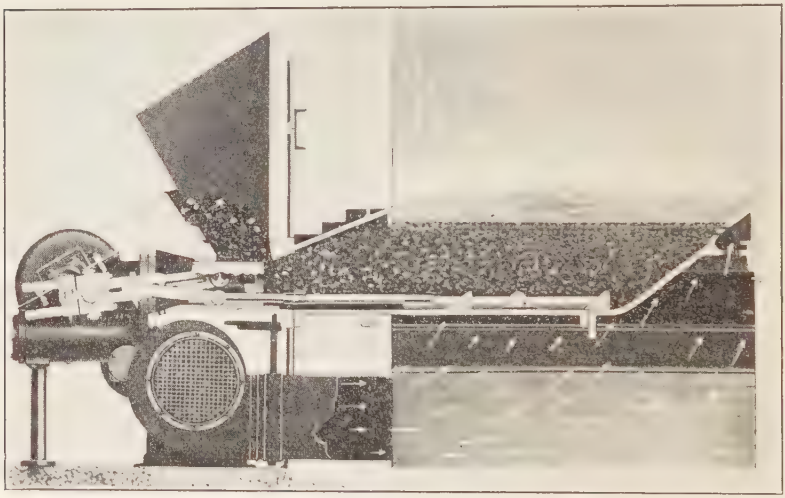


Fig. 64.—Longitudinal section of Detroit unistoker, showing adjustable plunger feed. Arrows indicate flow of air from fan to fuel bed.

including that which lies on the ash dump sections. The ash is discharged to the pits beneath, where it is quenched and cooled by water from the ash spray pipes, before removal through large doors at the front of the furnace at boiler-room floor level. This stoker is also made in the multiple-retort type.

The Detroit unistoker, shown by Figs. 64 and 65, combines the electric motor, air blower, and fuel-feeding mechanism into one unit mounted at the front of the stoker. The blower discharges directly into the air chamber, thus eliminating long air ducts.

The Taylor stoker, as illustrated by Figs. 66, 67, and 68, is typical of the modern inclined multiple-retort underfeed stoker. It consists of alternate retorts and tuyère boxes. The entire bottom of each

retort is formed by a series of overlapping and interlocking (adjustable) distributing rams, driven by the main feeding plunger, as shown in

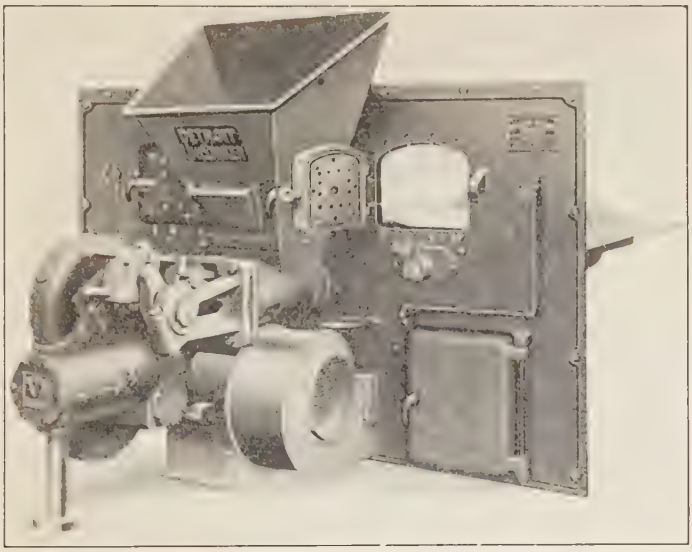


FIG. 65.—Front view of Detroit unistoker.

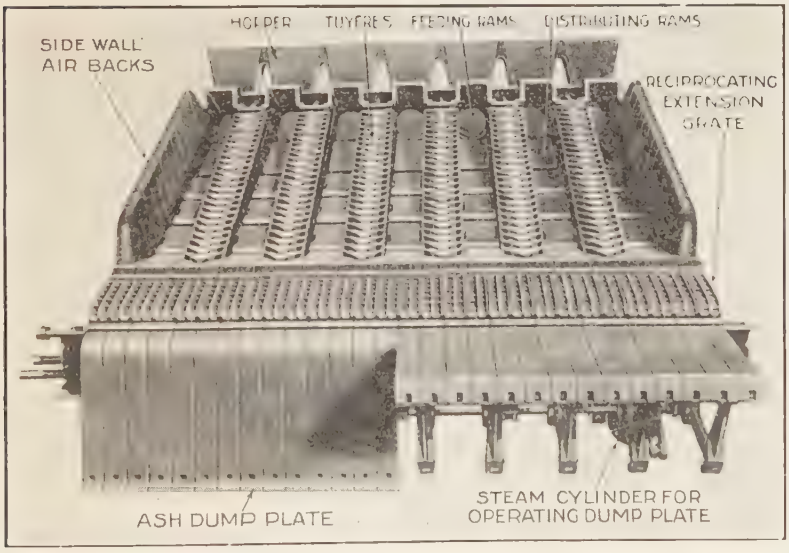


FIG. 66.—Furnace view of Taylor underfeed stoker.

Figs. 67 and 68. The volume of each retort can be varied by changing the position of the distributing rams. The coal is forced into each retort

by a reciprocating ram or plunger. When this plunger has moved about two-thirds of its stroke (this distance is adjustable) it starts to drive the distributing rams; during the remainder of the stroke the main and distributing rams move together, carrying the entire fuel bed with them. Because of the sloping faces of the distributing rams the fuel bed is raised upward as it is pushed toward the end of the retorts. The movement is small and gradual, and it has a mild slicing action which aids combustion. The fuel bed, being tightly packed, remains in place when the distributing rams are withdrawn on their outward stroke, leaving a small space along the entire under side. Green coal is forced down into this space during the first part of each inward stroke of the feeding plunger, before the distributing rams start to move.

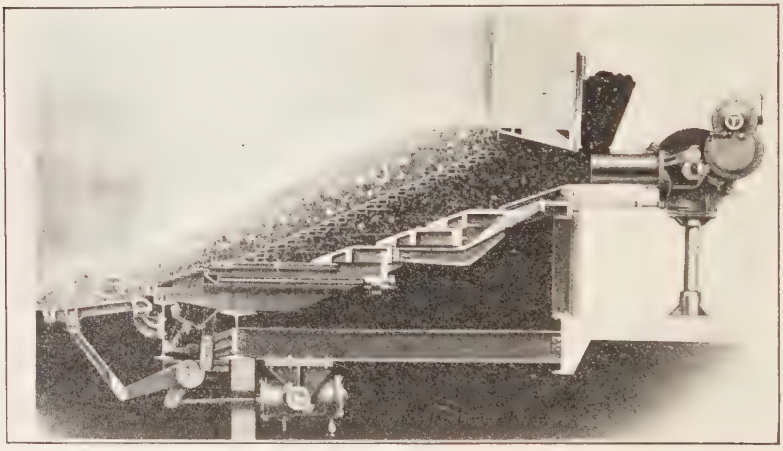


FIG. 67.—Longitudinal section of Taylor underfeed stoker with steam-operated ash dump.

Step by step, the coal is pushed up into the coking zone where it is gradually coked by the heat of the fire above. The volatile gases released pass upward through the burning fuel. Then the fuel is forced up into the zone of combustion where it forms the bed of incandescent coke and works slowly downward toward the ashpit, aided by the action of gravity and the distributing mechanism. To burn out any carbon remaining in the ash after it passes out beyond the retort section, a movable overfeed extension grate is provided. This extension grate is connected to the lower distributing rams as shown in Figs. 67 and 68, and in this way is given a reciprocating motion that is coordinated with the rest of the distributing system in moving the lower portion of the fuel bed.

Ash disposal is accomplished either by a hand- or steam-operated

dump plate (Fig. 67) or by an ash hopper and crusher rolls (Fig. 68). The dump plate is arranged to swing above the horizontal to clean the bridge wall of clinker adhesions. For large capacities and where loads are nearly uniform, the rotary ash discharge is applicable, but where

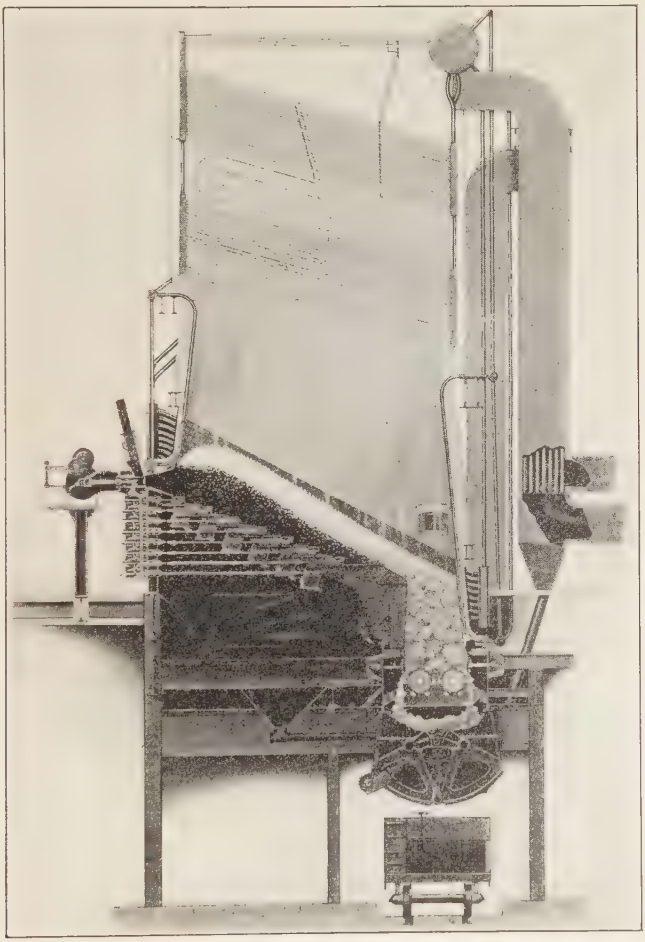


Fig. 68.—High-capacity Taylor stoker with water walls, ash hopper, and crusher rolls.

extreme flexibility is desired the dump plate is preferable. The area under the entire fuel-bed-supporting structure is enclosed; air under pressure is supplied to this space and is introduced at the rear. This arrangement gives the maximum cooling effect on that part of the grate which is subjected to the greatest heat and preheats the air before it

passes through the tuyère box openings into the furnace. Air is also supplied to the fuel through openings in the distributing blocks.

Each distributing ram and each section of the extension grate in Fig. 68 is driven by rods brought out to the front of the stoker, where the stroke of each may be individually controlled from the operating floor. The ash hopper, Fig. 68, is especially designed for use with long stokers of large fuel-burning capacity. This hopper is wider at the bottom than at the top, a feature which prevents arching of the clinkers. The depth is ample to retain the ashes long enough to thoroughly burn out the combustible matter and cool the ash for crushing. Air for these purposes is admitted through ports in the front wall of the ash hopper.

Underfeed stokers are particularly well adapted for coals high in volatile constituents which have caking characteristics and the ash content of which is low and non-clinkering. The continual agitation of the fuel bed keeps it porous even with coals that cake. Where quick steaming is desired to meet high overloads, the underfeed stoker may be used to advantage. Eastern coals may be burned continuously at a rate of 60 lb. per sq. ft. of grate surface, per hour, and for short periods of time the combustion rate may be increased to 70 lb. With western coals, the combustion rate is lowered to around 40 lb. Fires on underfeed stokers are easily banked and may be brought into service again in a short time.

104. Pulverized Fuel Equipment.—Steam generators using pulverized fuel require equipment to convert the raw fuel (coal, coke breeze, or lignite) into a powder and convey it to the furnace, into which it is delivered as required and burned in suspension in an atmosphere of air, similar to the burning of a gas. Correct furnace design and adequate fineness of pulverization, two important factors in the successful use of this fuel, were developed slowly. Previous to the year 1920, pulverized-fuel burning in boiler plants was in the experimental stage; since that time the increase in its use has been very rapid and today (1933) about one-fourth of the steam generated by coal-fired furnaces is being produced by pulverized fuel. There are boiler units fired with stokers generating 500,000 lb. of steam per hour; however, under some conditions pulverized-fuel firing has advantages, and this type of furnace has been used for practically all of the large steam-generating units recently installed which use coal for fuel.

Two systems are used to prepare and burn pulverized fuels: (1) the *central* or *storage system*, with its independent preparation plant, pulverized-fuel transport system, pulverized-fuel bins, and feeders; (2) the *unit* or *direct-fired system*, in which one machine (pulverizer) prepares and delivers the fuel to the furnace with some of the air necessary for combustion. The storage system was the first adopted; a survey of

recent installations shows a marked tendency to unit systems (mill drying) regardless of the kind of solid fuel used. Inasmuch as the

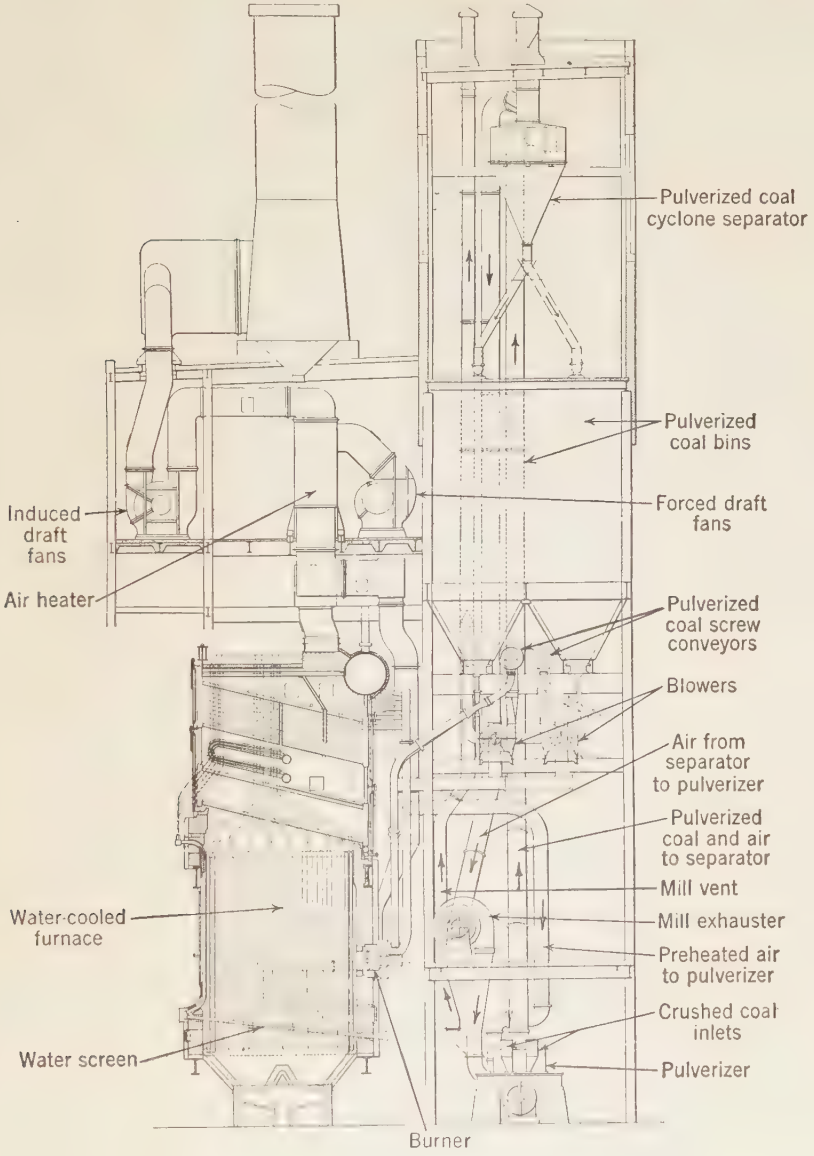


FIG. 69.—Storage type pulverized-fuel system.

efficiency of unit systems is as good as that obtained with storage systems and since improvements in pulverizers have rendered them as

reliable as the rest of the installation, the saving in investment assures the increased use of unit systems. The elimination of the numerous auxiliaries necessary with the storage system has made the simplicity of unit system installations their greatest advantage; other advantages include low first cost, lower cost of operation, and decreased maintenance.

With the storage system for burning pulverized fuel, Fig. 69, the usual equipment includes a raw coal crusher, magnetic separator, a

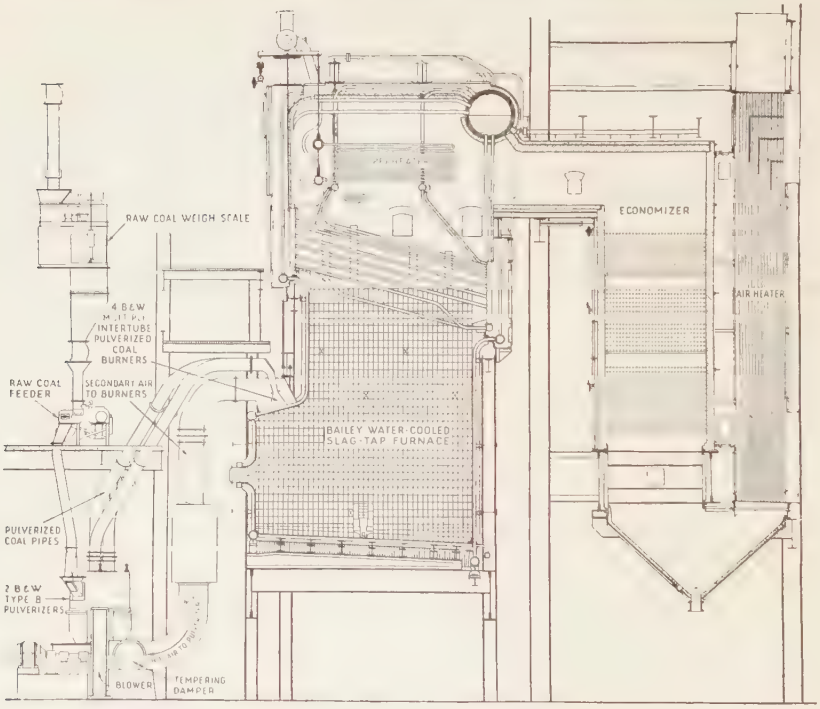


Fig. 70.—Direct-fired pulverized-fuel unit; cross-drum boiler with superheater, economizer, and air heater.

dryer for removing moisture from the fuel, a pulverizer for grinding the fuel to the required fineness, a screw conveyor or blower system for transporting the pulverized fuel to the boiler room, a blower for feeding the fuel to the burners, and the burners. Some fuels are free enough from moisture so that they can be pulverized and fed to the boiler without drying, in which cases the dryer would be omitted from the equipment required. Most of the recent installations have abandoned separate dryers in favor of drying by the use of preheated air in the pulverizer.

The direct-fired system, Fig. 70, reduces the pulverizing equipment to a minimum; a complete pulverized fuel system is provided for each

boiler. The fuel is used immediately after it is pulverized, thus avoiding any necessity for storage; the transportation of the pulverized fuel is simplified as the pulverizers are located near the boilers, and the use

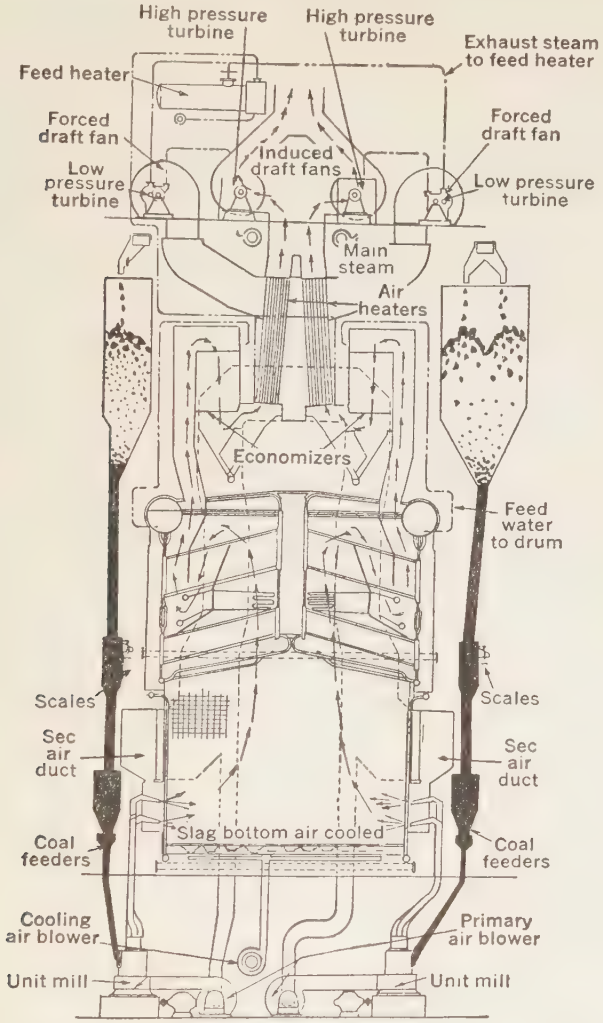


FIG. 71.—Cross-section of one of the million pound per hour steam generators at Hell Gate Station.

of preheated air through the pulverizers eliminates the necessity for separate dryers.

In each system the fuel is first crushed to a general size of $\frac{3}{4}$-in. to 1-in. lumps, and then passed over a magnetic separator to remove tramp

iron, such as bolts, nuts, rivets, etc. If this iron were not removed, damage would result to the pulverizers. Next the fuel goes to the pulverizers, and then either to a pulverized-fuel bin in the storage system, or directly to the boiler furnace in the direct-fired (unit) system.

The use of pulverized fuel has made possible large-capacity steam generators and higher efficiencies because of more complete combustion

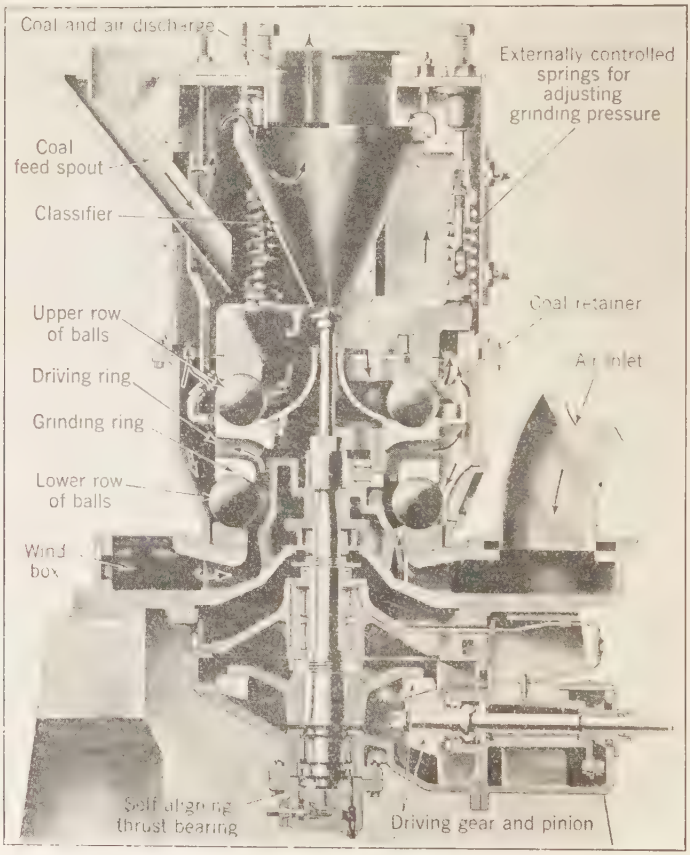


Fig. 72.—Cross-section of Babcock and Wilcox type-B pulverizer.

of the fuel. Figure 71 shows one of the largest boiler installations in the United States. It has generated over a million and a quarter pounds of steam per hour; this limitation of output was set by factors outside of the boiler and furnace, namely by the size of the feedwater piping.

105. Pulverizers: Firing Pulverized Fuel.—Pulverizers may be divided into three types, depending on the method of fuel reduction,

namely, *roller*, *impact*, and *ball mills*. The Babcock and Wilcox type B pulverizer, Fig. 72, is a roller mill embodying the spherical ball and grinding ring principle. Two rows of heavy steel balls are separated and propelled by a rotating ring, which is driven by and floats on the main driving shaft. The pressure between the balls and the rings is applied and kept constant by externally controlled steel springs.

In Fig. 72 crushed fuel enters at the upper left side of the pulverizer and enters the upper (stationary) grinding ring as indicated by the arrows. After being ground by the upper row of balls it passes through the annular opening formed by the intermediate (rotating) grinding ring and the coal basket. The coarser particles drop to the lower row of grinding balls for further pulverizing, while the finer particles are entrained by the air stream. The separating air enters at the lower right side of the pulverizer; this air moves upward inside and through the lower row of balls, then outward through the cored holes in the intermediate grinding ring. These changes in direction of the air stream tend to cause the coarse particles to readily separate from the fine particles. Separation is further assisted by an adjustable sleeve at the discharge of the mill. The pulverizer uses preheated air as a drying and separating medium, and it can be operated either as a pressure or suction mill to meet the requirements of the boiler installation. The unit is manufactured in capacities up to and including 40 tons of fuel per hour.

Impact pulverizers employ paddles or blades which rotate within a casing at a speed of 1000 to 2000 r.p.m. The impact of the blades and the attrition of the fuel particles on each other effect the desired fineness. Impact-type mills are used for capacities up to 10 tons of fuel per hour. The ball mill is used for the smaller units, less than 6 tons of fuel per hour; it consists of a horizontal steel drum containing a large number of small steel balls. In operation, the fuel enters one end of the conical drum and is crushed by the constant shifting of the balls as the drum rotates.

Crushed coal is fed to pulverizers at a desired rate, and after being pulverized it is removed by a flow of air produced or induced by a fan which, in some cases, is integral with the pulverizing mechanism. In direct-fired installations the primary air supplied to the furnace with the fuel is that which passes through the mill. The degree of fineness of the discharged fuel depends largely on the velocity of the air passing through the mill; common practice considers 75 to 80 per cent through a 200-mesh (openings per inch) and all through a 50-mesh sieve as the necessary fineness in most cases to prevent slagging troubles and to insure complete combustion.

Though fineness is highly favorable to combustion, other factors,

such as dryness and turbulence in burning, have significant effects. Turbulence and mixing of the fuel and air as they enter the furnace are effected by the burner design, velocity of the mixture leaving the burner, and the method of introducing the secondary air to the furnace.

There are three methods of firing pulverized fuel furnaces—*vertical*, *horizontal*, and *tangential*. The vertical type of burner is shown in Fig. 70, the horizontal in Fig. 71, and the tangential in Fig. 73. With vertical burners, the primary air carries the fuel and enters the furnace through the burner nozzle; secondary preheated air is admitted through the casing surrounding the burner; additional air may be admitted through the front wall of the furnace. In the horizontal and tangential burner arrangement the primary air carries the fuel and enters the furnace through the burner nozzle, and the secondary air enters either through dampers in the burner body or through the casing around the burner; there is no additional air supply to furnaces employing horizontal or tangential burners.

By the tangential method of firing, Fig. 73, is produced intensive turbulence of the burning mixture and a rotative motion so that the mixture completely fills the furnace, thus making all of the combustion space effective. This makes it possible to operate the furnace at high average rates of heat liberation with small loss from incomplete combustion. From one to four burners, stacked above each other, are located in each corner. Installations have been made which demonstrate that this method of firing is also suitable for gaseous and liquid fuels and that it permits the burning of any combination of these fuels simultaneously in the same furnace. The tangential method of firing is well adapted to use where high capacities are desired and where floor space is at a premium. This layout simplifies the arrangement of air ducts and feed pipes to the burners, as shown in Fig. 73.

Advantages claimed for the use of pulverized fuel include: complete combustion; simpler ash disposal; absence of smoke; use of cheaper fuels; ease of shift to use other solid fuels, oil, or gas; easy control of fuel and air supply; simplification of equipment and unification of design of boiler and furnace; absence of all moving parts in the furnace; no standby losses; low labor cost; increase of capacity per unit of furnace volume; and generally higher efficiency. Some of the disadvantages are: high first cost, cost of fuel preparation, storage limitations of pulverized fuel because of caking or fire hazard, furnace depreciation, and the possibility of refuse discharging through the chimney.

106. Water Walls and Water Screens.—A pulverized-fuel furnace should be designed so that any kind of fuel may be burned efficiently and at the required rates of combustion, without smoke and undue maintenance cost. The volume of the furnace depends primarily upon

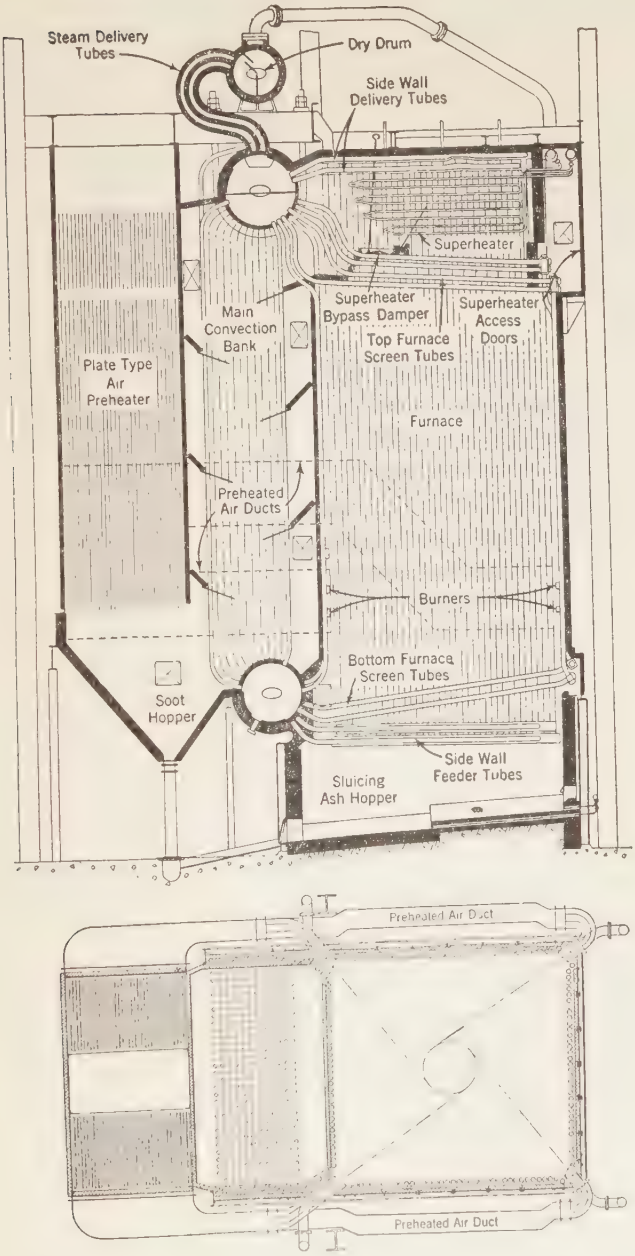


FIG. 73.—Combustion steam generator showing corner-tangential firing.

the rate of combustion, fineness of the fuel particles, and the method of introducing the air and fuel into the furnace. These are coordinated with the type of wall construction and the rate of heat liberation; 50 to 75 cu. ft. of furnace volume per lb. of fuel burned per minute are average values used. The high efficiency obtained with the use of pulverized fuel is due to the small amount of excess air; this means high furnace temperatures and erosion of the ordinary refractory walls when using a fuel having a low-temperature fusing ash.

The temperature of the furnace lining may be maintained below that of the slag by using water screen tubes, Fig. 73, or by placing a radiant-heat superheater, Fig. 83, in one or more of the brick walls. Air-cooled walls, water tubes in front of the walls, and recessed water tube refractory-faced walls are also being used. The *air-cooled* type of *wall* was used in some of the first pulverized-fuel furnaces. An air space was left between the firebrick lining of the furnace and the outside wall; air was circulated through this space, heated, and admitted to the furnace. Experiments showed that better results could be obtained by lining the walls with water tubes and placing a water screen near the bottom of the furnace. The *water screen*, Fig. 73, consists of horizontal rows of water tubes spaced far enough apart so that ash or slag can readily drop between them. In falling through this screen, the ash particles are cooled and solidification occurs before the ash reaches the bottom of the furnace. However, at heavy loads the flame may penetrate the screen and slagging on the tubes and in the ashpit may result.

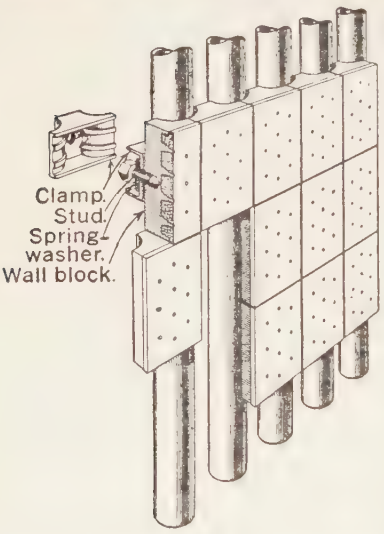


Fig. 74. Details of Bailey water-cooled furnace wall.

Frequently in stoker-fired and pulverized-fuel installations the inside of the front and rear furnace walls is lined with water tubes directly exposed to the heat of the furnace; see Figs. 68, 69, 70, 71, and 73. The water tubes which constitute the walls and water screen are connected with the boiler. They add to the heat-absorbing capacity of the boiler and are cheaper to maintain than refractory walls. Details of the Bailey *water-cooled wall* are shown by Fig. 74; this is a recessed water tube refractory-faced wall. The water tubes are covered with

very closely fitting cast-iron blocks. These blocks are held to the tubes by means of threaded studs and special clamps. The opposite sides of the blocks are recessed to hold firebrick tiles, which present a smooth vertical refractory surface to the interior of the furnace; see also Fig. 70. Near the top and bottom of the furnace, where the heat is not very intense, bare iron blocks are generally used.

The successful collection and disposition of the ash from burning pulverized fuel constitute another problem. About 50 per cent of the ash from pulverized fuel is carried through the boiler-passes and out the stack; and that which remains is deposited at the bottom of the furnace, whence it is removed at intervals. In *slag-tap furnaces* the ash

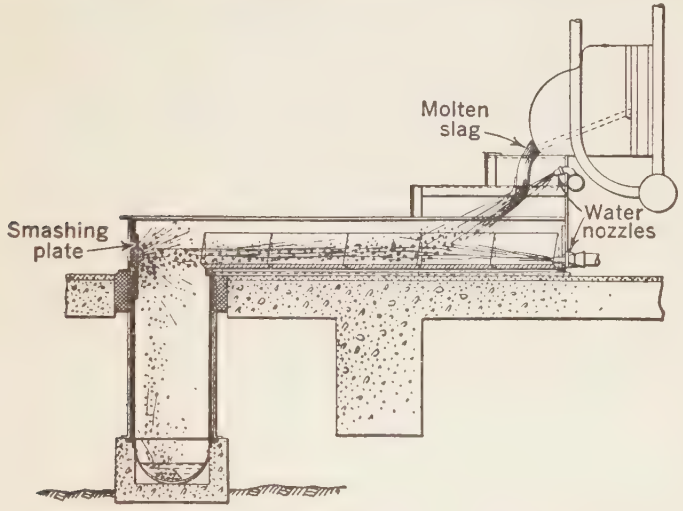


Fig. 75.—Details of sluicing molten slag from a pulverized-fuel furnace.

is allowed to remain in a molten state, in which case it is drawn off, cooled and broken up by water jets, and carried away (sluiced) by water, as shown by Fig. 75. This method is especially applicable where the ash from the fuel used has a low fusing temperature, below 2500 deg. F. Some of its advantages are: requires a minimum of building volume, will burn efficiently and at high ratings an unusually wide variety of fuels, produces quick ignition of the fuel, prevents infiltration of air into the bottom of the furnace, reduces ash-handling labor, and makes ash removal remarkably free from dust and dirt. The solid or granular ash is removed by the ordinary means as used with stokers. The elimination of ash discharge into the atmosphere has in some instances become a decided civic problem, and legislation has been passed regulating its control. Many comprehensive efforts have been made to

collect this fly ash by the use of electric precipitation, water spraying, and mechanical separation in baffle and cyclone collectors.

107. Coal and Ash Handling.—In the small plant, less than 500 hp., coal and ashes frequently are handled by means of shovel and wheelbarrow. Larger plants are generally equipped with conveying machinery to insure continuity of operation and to dispense with unreliable labor. The method of delivering the fuel to the furnace and of conveying the ashes away should be one that will give good results for minimum cost. Some of the methods used are: (*a*) wheelbarrow and hand shoveling, (*b*) continuous conveyors, endless belt and pivoted buckets, (*c*) pneumatic as used for pulverized coal, (*d*) hoists and cranes, and (*e*) hydraulic sluice. Ashes are sometimes handled by the same

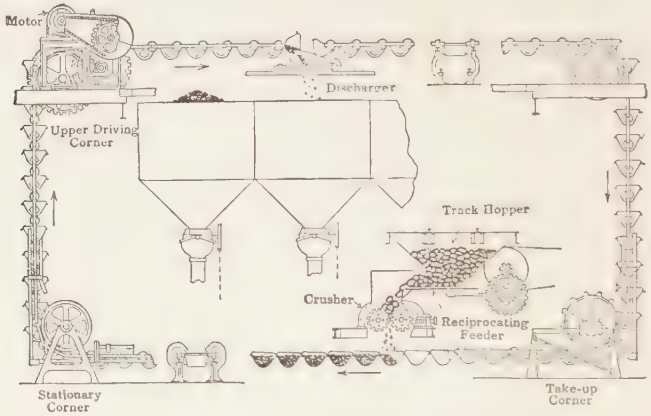


FIG. 76.—Peck pivoted-bucket conveyor.

conveying system used for the coal, but in many plants the disposal of each is taken care of separately because of greater flexibility of operation, and on account of the abrasive and corrosive action of the ashes. Figure 76 shows a diagrammatic arrangement of a pivoted-bucket conveyor which is one of the popular methods for handling coal from car to furnace. The buckets, hung on pivots and suspended between two endless chains, maintain their upright position at all times. The chain is driven by a motor and sprocket wheels. The point of delivery of the coal is controlled by the location of the movable discharger.

108. Oil- and Gas-burning Equipment. - The application of improved and varied fuel-burning equipment has intensified the competition between solid, liquid, and gaseous fuels. Increased oil and natural gas production have gone hand in hand, for geological reasons, and petroleum is now the second largest producer of energy (about 30 per cent of

the total) in the United States. It ranks next to coal, and during the last decade oil and gas have replaced coal to a large extent.

Where available for steam-generation purposes, fuel oil has the following advantages: low cost for equipment and installation, convenience of storage, ease of handling, flexibility of operation, small labor cost, cleanliness, ash-free and smokeless combustion; also, it usually gives increased steam capacity and higher boiler efficiencies than coal. In many localities, oil is more expensive than coal, so that, despite the many advantages named above, the net operating costs with fuel oil may be too high to warrant its use.

The name *oil burner* is a misnomer, as the so-called burner does not burn the oil but merely atomizes it. The atomization may be effected

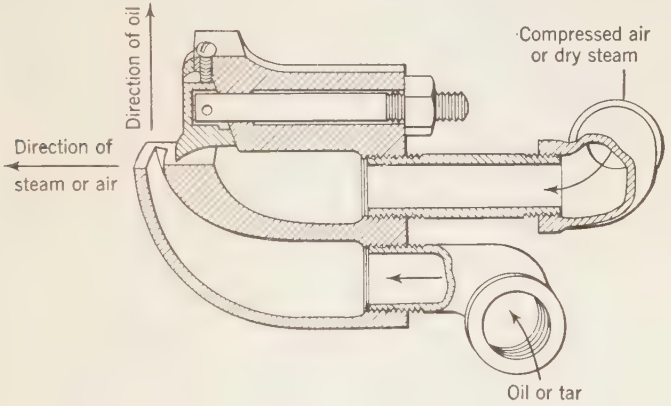


Fig. 77.—Best oil burner, air or steam atomization.

by high- or low-pressure steam, gas, or compressed air, a combination of steam and air, or by mechanical means. The four basic requirements for the economical use of fluid fuels are: atomization, adequate air supply, effective distribution of the heat, and efficient operation and maintenance of the furnace. The fineness of the fuel spray must be sufficient to permit intimate contact with the air for combustion. It is of equal importance that the air be supplied in such manner and such quantity that the intimate mixture will be made at the entrance to the furnace rather than later in the furnace itself. The design of the furnace and the heat passages through the boiler will affect the overall efficiency. The furnace must have sufficient volume; its brickwork must be so placed as to aid in securing best efficiency of combustion; and, at the same time, operating difficulties and maintenance must be kept at a minimum.

Steam atomizing oil burners may be designated as external atomizers or internal atomizers. Figure 77 shows a section through the Best

burner, illustrating a well-known design of the external-atomizing type. The oil is supplied at a pressure of less than 25 lb. per sq. in. through the lower pipe at right angles to the high-velocity steam jet (in every case where steam is mentioned, compressed air may be substituted) and is discharged in the shape of a flat or fan-tail spray. Steam or air is supplied at pressures of 25 to 125 lb. per sq. in. This burner has been designed especially to eliminate the possibility of clogging or carbonizing and to insure thorough atomization and combustion. The Airoil burner, Fig. 78, is of the internal-atomizing type, having an enlarged chamber inside the burner into which oil and steam (or air) are supplied at high velocities through separate nozzles, in this chamber the oil and steam are thoroughly mixed before passing into the furnace. The

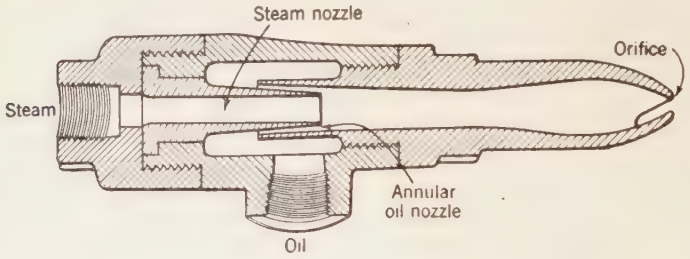


Fig. 78.—National "Airoil" steam-atomizing oil burner.

discharge orifice may be varied to give either a flat flame or round flame, as desired.

The centrifugal force required for *mechanical atomization* may be produced in either of two ways—by pressure or by mechanical rotation. The pressure type is preferable because in the mechanical rotation burner the maintenance cost of the moving parts is high as they are exposed to the radiant heat of the furnace. Figure 79 gives the general details of the Peabody "Wide-Range" mechanical atomizer, pressure type. Oil is supplied at a pressure of 200 lb. per sq. in. into the annular space between the barrel of the atomizer and the inner tube. After flowing through the outer series of holes in the tip plug, the oil enters the annular groove in the tip and passes through the tangential slots which impart the rapid rotary motion to the oil in the central whirling chamber of the tip. A portion of the oil is then sprayed through the orifice into the furnace (the actual proportion depending on the setting of the master control valve on the return line), and the remainder passes through the inner series of holes in the tip plug into the inner tube. The Peabody mechanical oil burner assembly of Fig. 80 shows the air register, atomizer, and fuel lines. Oil is supplied to the burner through the right-hand flexible hose; from the inner tube the return oil flows back through

the left-hand flexible hose to the return oil header. By connecting all the oil return lines to a common return header provided with a master control valve, the burners may be simultaneously adjusted for any capacity desired.

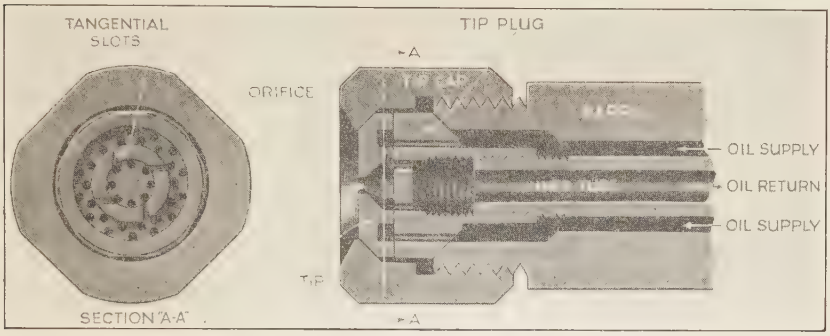


Fig. 79.—Details of Peabody "Wide-Range" mechanical atomizer.

Steam atomizers for boiler furnaces operating with natural draft are usually designed to give a flat flame and are installed in the furnace so that the flame is spread out over a floor of refractory material. When they are provided with orifices which produce a hollow conical jet, air

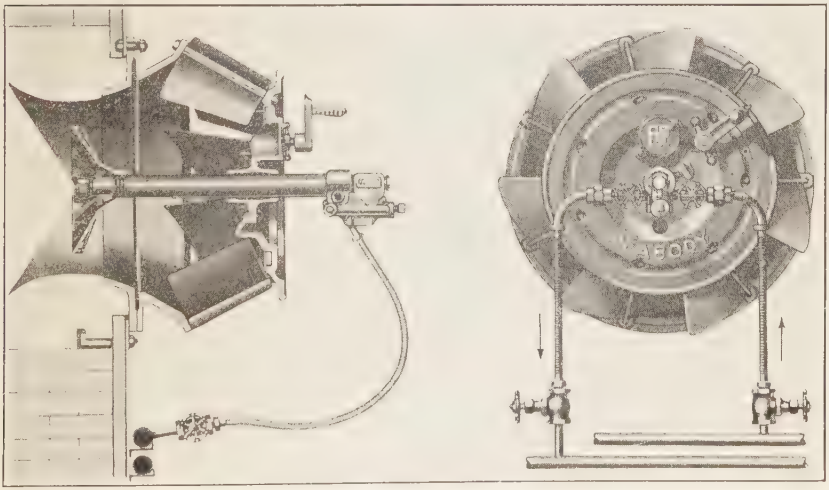


Fig. 80.—Peabody mechanical oil burner assembly, showing air register and atomizer.

registers, as shown in Fig. 80, are necessary for efficient mixing of the air and atomized fuel. Steam atomizers in connection with air registers are also suitable for forced draft. Mechanical atomizers are always provided with air registers which are usually arranged for forced draft.

Air registers are fully as important from the standpoint of design and construction, and have as great an influence upon the obtaining of perfect combustion, as are the atomizers.

Steam is an effective but uneconomical atomizing agent. It is possible under ideal test conditions to operate steam atomizers with a steam consumption of 0.25 lb. of steam per lb. of oil fired, or approximately 2 per cent of the boiler output. Under operating conditions, the actual steam consumption ranges from 0.5 to 1.0 lb. of steam per lb. of oil, which corresponds to about 4 to 8 per cent of the steam output of the boiler. The equivalent steam consumption of the mechanical burner depends primarily upon the oil pressure maintained, the efficiency of the oil-pumping and heating apparatus, and whether or not the exhaust steam is utilized. In general, the steam consumption for mechanical atomizer operation will not exceed one per cent of the total steam generated.

The advantages of steam atomizers as compared to mechanical atomizers are: ease of manufacture, simplicity of installation, low oil pressure (seldom more than 30 lb.), flexibility and ease of operation, high boiler efficiencies, safety due to low pressure and temperature of oil, and flexibility of shaping the flame. Some of the disadvantages include: cost of steam used for atomizing, loss of boiler water, noise of operation, frequent cleaning of burners, blow-pipe action of the flame in small combustion chambers, heat loss in the flue gases, and extra amount of moisture in the flue gases. Steam atomizers are seldom designed to pass more than 1000 gallons of oil per hour per burner, whereas single mechanical burners have satisfactorily handled as much as 5000 gallons per hour.

Air atomization is restricted to low capacities on account of the high cost of air compression. Natural gas and by-product gases are sometimes used as atomizing agents at pressures of 50 lb. or higher. Gas atomization assists in maintaining ignition in burning special fuels and is often found to be very economical as the heat in the gas can be utilized in the furnace. *Combined burner units* are now available for burning gas or oil in addition to pulverized fuel without sacrificing any of the advantages of each individual burner. The advantages of a combined burner are primarily a standby reserve of fuel and the ability to use the most economical fuel under varying market conditions. Other advantages include: the construction and operation are simple, because of fewer burners; all burners may be in operation at all times, thus eliminating necessity for protecting idle burners; continuity of operation is assured; two fuels may be burned simultaneously in the same burner if desired; and first cost and maintenance costs for burners and furnace are lower.

The widespread extension of pipelines for the transmission of *natural gas* and recent developments in the natural-gas industry have removed in a large measure the restricted territorial use of this fuel. Small industrial plants which have isolated steam units may consider gas as the most economical fuel. Gas has all the advantages of other fuels without any of the disavantages, hence the primary cost is often of secondary importance. The boiler efficiency with oil is generally a few per cent higher than with gas owing to higher " hydrogen losses " with gaseous fuel. However, the use of gas has proved economical because of the cost of oil atomization and steam required for soot blowing and heating the oil in the tanks; also, handling and maintenance costs are higher for oil than for gas. Superheated-steam temperatures are considerably higher when burning natural gas than when burning oil, owing to lower water-wall absorption and larger flue-gas volume while burning gas. Natural gas is the most popular gaseous fuel, but a number of steam-generating units are equipped for burning coke-oven or blast-furnace gas, which are by-product gases of the iron and steel industry.

Furnaces are generally equipped to burn gas simultaneously with either oil or pulverized fuel, or to burn gas only as a standby fuel. The volume of the combustion chamber in recent gas- and oil-fired plants with high settings ranges from 0.3 to 0.7 cu. ft. per sq. ft. of boiler heating surface or from 0.1 to 0.3 cu. ft. per sq. ft. of total heating surface. The best results have been obtained where the flame travel is complete without impingement on tubes, walls, or floors, and where as much of the boiler heating surface as possible is exposed to radiant heat. In general, the larger the furnace and the farther the burners are located from the tubes or heating surfaces, the better will be the results obtained and the greater the overload capacity of the steam generator.

The design of gas burners is largely dependent on the kind of gas to be used. The gas and the air for combustion may be mixed inside the burner jet, but most gas burners are of the outside mixing type and admit air to the furnace outside of and around the gas burner jet, like the mechanical burner shown in Fig. 80. The air for combustion is drawn (or forced) through the air register and is given a whirling motion by the setting and shape of the vanes so that both air and gas are thoroughly mixed as they enter the furnace.

109. Air Heaters. As used with a steam generator, the purpose of an air heater is to extract heat from the flue gases and transfer it to the air used for combustion of the fuel. The greatest source of loss in connection with steam generation is the quantity of heat carried away by the flue gases. The forcing of boilers to high overloads and the desirability of complete combustion with a minimum air supply results in high temperatures of the exit gases. Economizers, described in Chapter IX,

have been installed in most plants of any importance in order to utilize some of the heat in the exit flue gases. The use of exhaust steam or steam from a bleeder turbine for heating the feedwater may eliminate to some extent the necessity of using an economizer, and as an alternative an air heater may be used. Advantages derived from the use of preheated air for combustion are: heat conservation, improved combustion, successful burning of low-grade fuel, increased efficiency, and probable increased capacity. The rate of ignition and combustion is greatly accelerated by the use of preheated air, making possible higher combustion rates per unit of furnace volume and, therefore, smaller furnaces. The hotter air supply increases the furnace temperature and hence increases the transmission of radiant heat to the boiler.

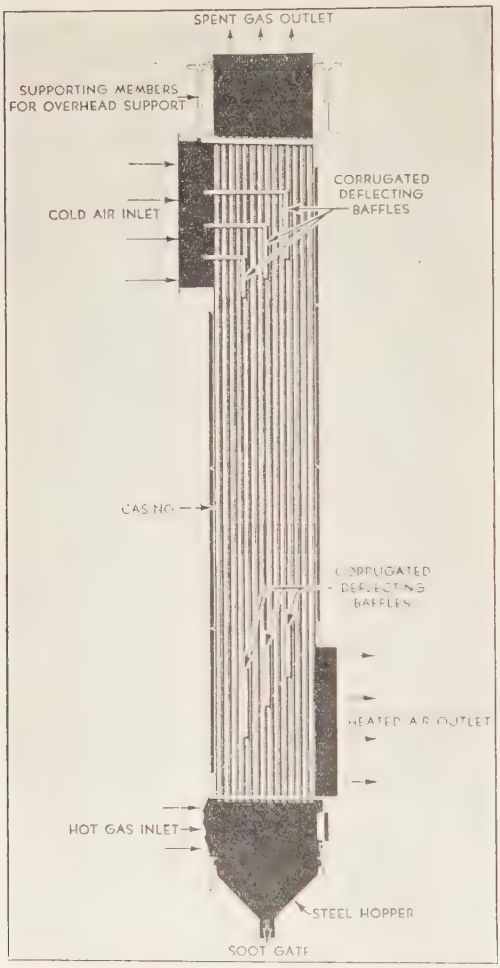


FIG. 81.—Babcock and Wilcox tubular air heater.

Air heaters use either the convection or regeneration principle of heat transmission. The *convection* type may be subdivided as plate or tubular, whereas the regenerative type employs a metal rotor which is alternately heated by the gases and then cooled by the air. Figure 81 shows a side elevation of a tubular-type air heater using the counterflow principle of heat transmission. The hot gases enter the tubes at the bottom and leave at the top of the heater as indicated. The cold air enters at the upper left side, flows around the outside of the tubes, and leaves at the lower right. Baffles are supplied as shown in Fig. 81 for the purpose of directing the flow of air so that the maximum heat

transfer is obtained; the baffles are arranged to keep the velocity of the air as nearly constant as possible. In the plate type of heater the gas and air travel through alternate passages between the plates.

The *regenerative* type of air heater is illustrated by Fig. 82. Fresh air is drawn by a fan into the upper portion of the casing which is divided into two parts by a vertical partition. The air is confined to one side of the partition and passes downward through a porous cylindrical drum or rotor to a semicircular chamber in the lower part of the housing. The flue gases from the boiler traverse the apparatus in a reverse path. They enter the lower semicircular chamber as indicated and pass upward through the drum rotor to the upper chamber, whence they are

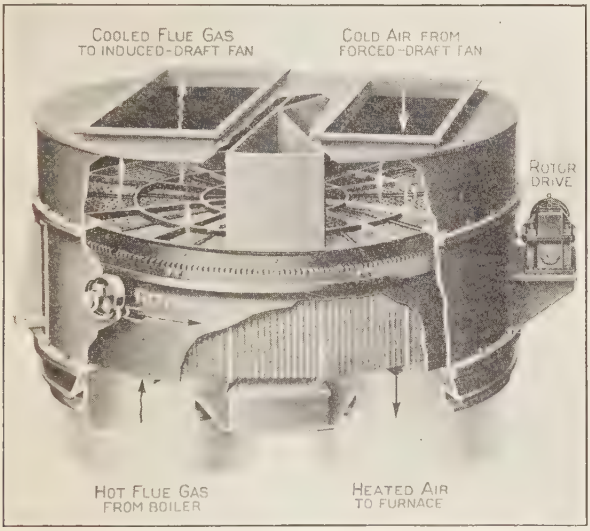


Fig. 82.—Ljungström regenerative-type air heater.

exhausted and impelled to the chimney by a second fan. The drum is kept in continuous slow rotation so that the part heated by the flue gases is constantly passing to the other side of the device and gives up its heat to the cold air sweeping through it. Similarly, the cool part of the rotor is returning to be reheated by the flue gases. Cleaning and corrosion are important factors affecting the performance of air heaters, especially the convection type.

110. Superheaters.—Superheated steam has a temperature higher than that of saturated steam at a corresponding pressure. Superheated steam greatly improves the economy of steam turbines and reciprocating steam engines, and is used extensively in large modern power plants. The increased economy may be effected in general by

increased overall boiler efficiency, reduced condensation losses in steam pipelines, reducing steam consumption of the prime movers, and increasing the capacity of the plant. The purpose of a *superheater* is to impart

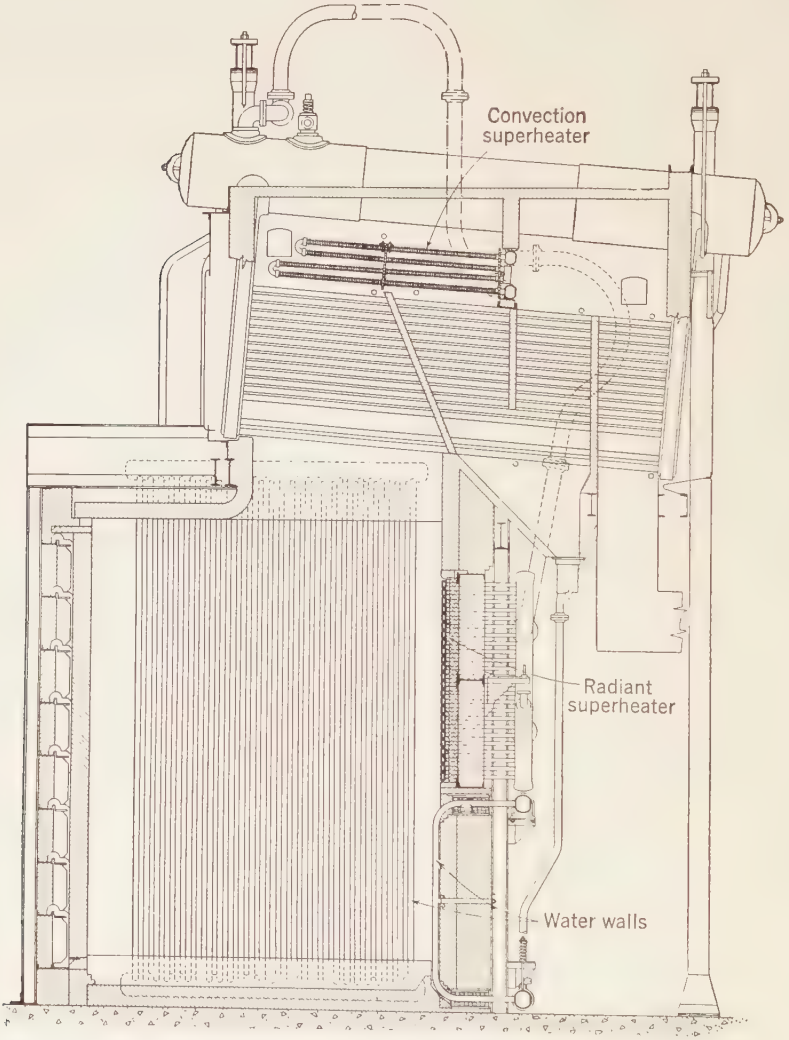


FIG. 83.—Convection superheater and radiant superheater.

additional heat to steam after it has left the boiler in which it was generated, thus increasing the steam temperature. The essential requirements of a successful superheater should include safety in operation, accessibility, protection against overheating, improved efficiency of the

combined boiler and superheater, regulation, provision for keeping units clean, and allowance for expansion and contraction of all parts.

Superheaters may be classified according to the manner in which the heat is transferred to the steam, as *convection* and *radiant* types. Convection superheaters may be independently fired or integrally built within the boiler setting. The independently fired superheater is not widely used, being confined in general to special and limited cases where a small amount of superheated steam is desired at a high temperature. The advantages of regulation and making repairs without affecting boiler operation are offset by higher fuel cost, lower economy, separate attention, and additional floor space required. Integrally built convection-type superheaters receive heat from the same flue gases and same

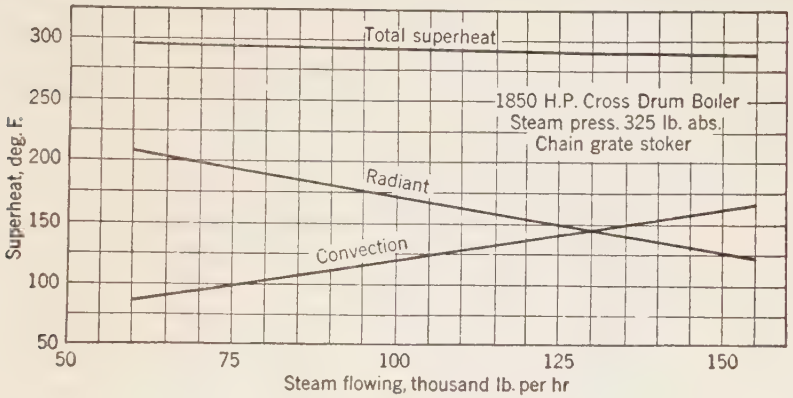


Fig. 84.—Typical performance curves of a combination superheater.

furnace as the boiler. They may be located among the banks of the boiler tubes, Figs. 37 and 40, or at the top of the tubes and below the drum, Fig. 83. The radiant superheater, Fig. 83, is placed on the furnace walls and absorbs heat by radiation from the fire. With convection superheaters the temperature of the superheated steam increases with an increase in the boiler capacity; see Fig. 84. Radiant superheaters have the opposite characteristic, i.e., the superheat decreases as the boiler output increases. Practically constant superheat is obtained at all loads, Fig. 84, by a combination superheater, part of the surface being a convection superheater connected in series with the other smaller part installed as a radiant superheater.

A radiant-heat superheater is shown in Fig. 83 installed in a pulverized-coal boiler setting. This superheater is placed to form a part of one of the walls of the furnace for the absorption of heat by direct radiation from the fire. The inlet steam header is at the top and the outlet superheater header is located at the bottom. Radiant super-

heaters serve as a protection for the furnace wall to which they are fastened, and high superheats are usually obtained. Radiation to the superheater depends on: (1) temperature, position, and radiating properties of the flame, (2) combustion rate, (3) excess air furnished, and (4) heating surface of the superheater.

Figure 85 illustrates the application of an *interdeck* convection superheater to a cross-drum Babcock and Wilcox boiler. The steam collected in the dry pipe of the boiler drum passes downward to the lower manifold of the superheater. From there the flow is through the U-shaped steel tubes into the upper manifold, thence to the superheated steam

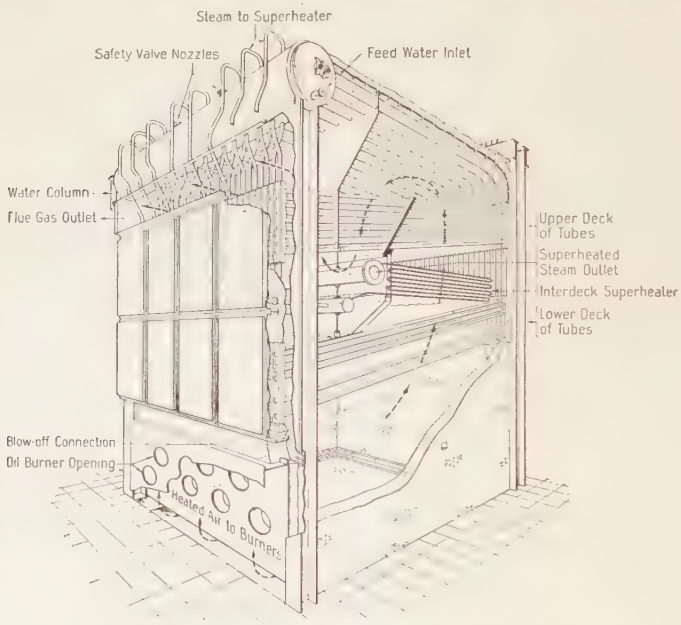


FIG. 85.—Interdeck convection-type superheater.

header serving the turbine or engine. For smaller boilers the convection superheater may be located above the boiler tubes between the first and second passes as shown in Fig. 83. However, more uniform superheat at various boiler outputs is obtained by placing the superheater between the tubes, Figs. 69, 73, and 85.

111. Desuperheaters. Industrial processes which require the use of saturated steam instead of superheated steam are: various chemical processes; the manufacture of sugar, paint, oil, textiles, cellulose, paper, etc.; and certain processes in connection with brewing, distilling, bleaching, and dyeing. In such plants there is a demand also for considerable power which can be generated more economically by the use

of superheated steam. It is therefore necessary to provide some means for furnishing both superheated and saturated steam in many plants. This could be obtained by using separate boilers, but, in general, this method is complicated and unsatisfactory. Practically all modern industrial plants are using the method of superheating all the steam and, by means of a desuperheater, removing the superheat from that portion of the steam required for certain equipment and processes.

Several types of desuperheaters are available, but the principle of operation is, in general, the same. The operation of desuperheaters involves the evaporation of a quantity of water equivalent to the reduction of the superheat of the incoming steam. The additional quantity of steam available at the exit, resulting from the evaporation of the water, compensates for the destroyed superheat, thus adjusting the equilibrium between the temperature and pressure of incoming and outgoing steam.

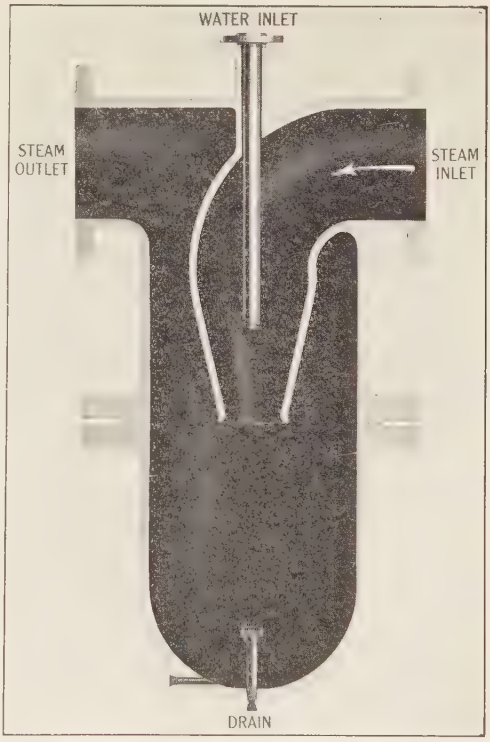


FIG. 86.—Swartwout spray-type desuperheater.

The principle of atomization as used in the automobile carburetor is applied in the spray-type desuperheater, shown by Fig. 86. It combines a spray nozzle, a cone, and a steam separator. Superheated steam enters as indicated by the arrow and passes into the cone. Here it is met by the cooling water which is introduced as a conical spray from the end of the water inlet pipe. The superheated steam, sweeping through the cone at a high velocity, instantly vaporizes the cooling water. The desuperheated (saturated) steam leaves as indicated; any entrained moisture is thrown out by the reversal of the steam flow and collects in the bottom of the desuperheater from which it may be removed by a steam trap.

112. Steam Separators and Traps.—Saturated steam flowing from the boiler to engines or turbines may contain water because of priming

in the boiler or condensation in the pipelines. With the small clearances now allowed in reciprocating engines, water in steam may break either the valve, cylinder head, or piston; in smaller but continuous quantities it may cause erosion of valves, valve seats, and cylinders. The use of wet steam also requires more cylinder oil and there is a constant loss in economy from increased initial condensation. With steam turbines, the moisture content reduces efficiency and causes erosion of the blades. The purpose of a live *steam separator* is to remove the water from saturated steam by the principle of reversed flow, baffle plates, centrifugal force, or by mesh separation. Figures 87 and 88 show sections through typical steam separators of the baffle type. The moisture is thrown out by the reversal of the steam flow or adheres to the surfaces and collects in the water chamber at the bottom of the separator, from which it may be removed by a steam trap. A *steam trap* is an automatically controlled valve, the function of which is to allow the discharge of separated water and to prevent the escape of steam. Steam separators, steam radiators or heating coils, steamjacketed kettles, or any steam apparatus from which it is desired to remove the condensate as rapidly as it accumulates should be drained by the use of a steam trap. Traps may be classified according to the principle of operation as float, bucket, bowl, and expansion. Figure 89

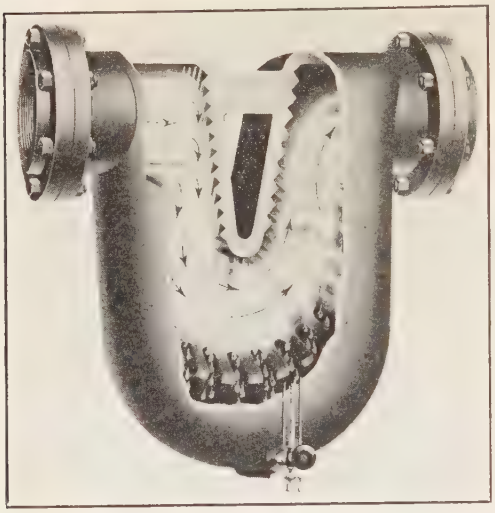


FIG. 87.—Wright-Austin horizontal steam separator.

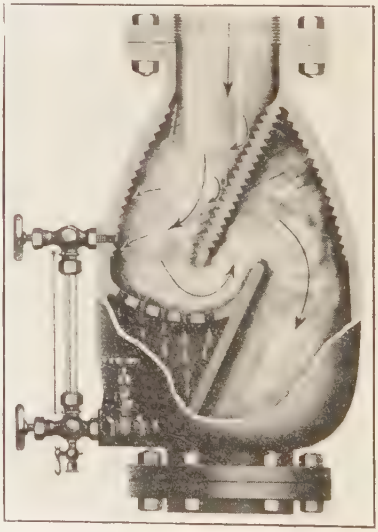


FIG. 88.—Wright-Austin vertical steam separator.

shows a section through a typical float trap in which water gravitating through connection I accumulates in the trap. The movement of float C opens the valve F; thus the water is discharged through H by the steam pressure acting upon the surface of the liquid. Water is removed until the level in the trap lowers the float enough to close valve F. Part A is used for manually opening valve F; B is an air vent, and J a sediment-discharge connection.

A typical bucket trap is shown in section by Fig. 90. The water of condensation enters the body of the trap and fills the space around the bucket. The bucket then floats and forces the valve against its seat, as shown. When the water rises above the edges of the bucket, an over-

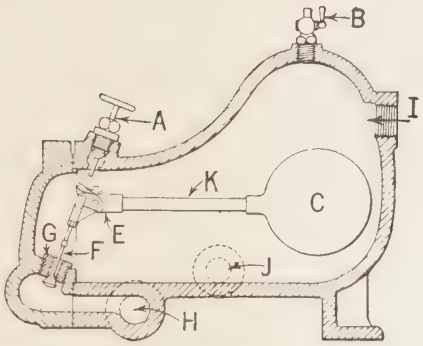


Fig. 89.—McDaniel float steam trap.

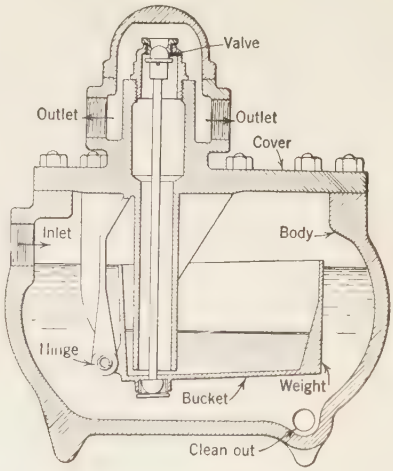


Fig. 90.—Squires bucket steam trap.

flow into the bucket causes it to sink, thereby withdrawing the valve from the seat. This permits the steam pressure acting on the water surface to force the water through the valve opening to the outlet connections. When the bucket is empty, it rises and closes the valve, and the cycle is repeated.

Figure 91 shows a section through an inverted bucket trap. The Armstrong trap utilizes the buoyancy of a floating bucket (when filled with steam) to close the valve, and the weight of the bucket (when filled with water) to open the valve. By the use of an ingenious floating lever of the valve mechanism in this trap the weight of the bucket is multiplied a number of times for opening the valve quickly, and the buoyancy without any leverage is utilized for closing the valve. Thus, this design produces a large force to open the valve against the pressure

in the line, and produces only a small force on the closing movement, thereby minimizing excessive wear on the entire mechanism.

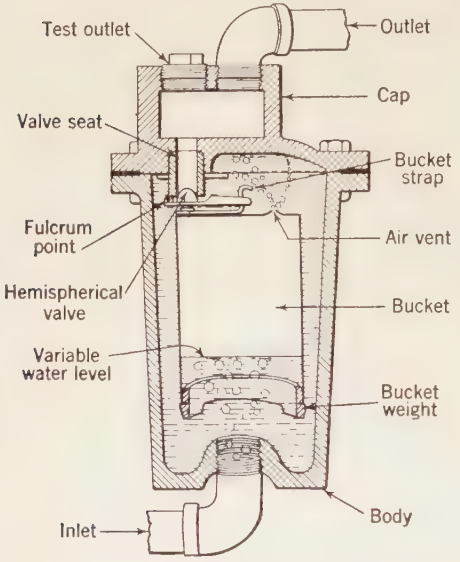


FIG. 91.—Armstrong inverted bucket steam trap.

113. Steam Purifiers.—All boilers, regardless of design, may deliver steam containing moisture, especially if fed with water that foams upon boiling. The percentage of moisture so delivered will depend primarily upon the rating being developed, the water level in the boiler, and the amount of impurities in the boiler water. The steam purifier is a separator installed inside the boiler drum in the same relative position as the dry pipe and so constructed that moisture and solids are removed as the steam passes through the purifier. The removal of impurities is desirable because of the resultant difficulties that may arise from deposits in the pipelines, traps, superheaters, valves, and prime movers. The maintenance of clean superheating surfaces permits better heat transfer and consequently higher superheat. The use of clean, dry steam in a turbine aids in the maintenance of the correct contour of passages by reducing erosion and corrosion, and prevents the depositing of foreign materials on the nozzles and blades.

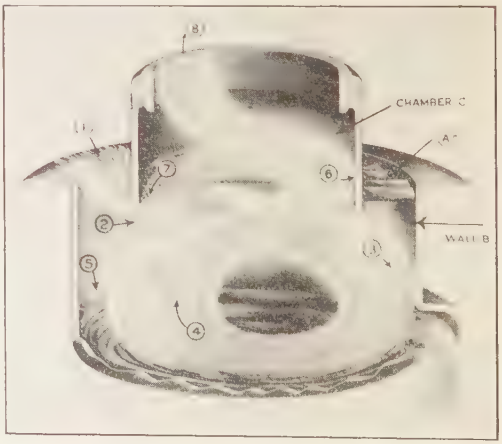


FIG. 92.—Hagan live steam purifier.

The Hagan steam purifier, Fig. 92, removes moisture and suspended solids from the steam by centrifugal separation. The steam to be cleaned enters the purifier through the directing vanes A, impinges against the walls of the separating chamber, and is given a whirling de-

scending and ascending motion as indicated by the arrows (in numerical sequence). The moisture is thrown against the wall B and is directed and gravitated toward the tangential discharge outlet, to a settling basin or blow-off. The steam reverses its direction and ascends as a revolving core surrounded by the revolving annular body of steam moving downward. Clean, dry steam leaves the purifier at the top (*S*) and is delivered to the steam header or superheater. "Tracyfier" is the name given to a steam purifier which consists of a number of rows of baffle plates so spaced that the steam is broken up into small, thin ribbons about $\frac{1}{20}$ in. in thickness. The moisture adheres to the baffles. The velocity through the channels is low so that the moisture is not picked up again by the steam; the moisture and solids gravitate to the bottom of the chamber and discharge to waste.

114. Steam Injectors.—An injector, Fig. 93, is a device consisting of nozzles so arranged that a jet of steam expanding therein attains a high velocity. The rapidly flowing steam meets a mass of water and condenses, and imparts sufficient momentum to the water to force it into the boiler. The steam flowing through nozzle S loses pressure and gains velocity as it enters the lifting tube V. A reduced pressure is created in suction pipe J, thus causing the water to rise until it comes in contact with the steam at the entrance to tube V. The steam loses velocity by impact and because the kinetic energy is absorbed by the water. It also shrinks in volume, owing to condensation, at the instant it meets the water and enters the lifting tube. Because of continued condensation, the mixture of steam and water continues to contract, and the lifting tube V and the combining tube C are gradually reduced in cross-section. When the combined jet of water and condensed steam reaches the delivery tube D, the passage is gradually enlarged. This provides space for the water at constantly decreasing velocity.

As the stream of water moves through the delivery tube, it is changing kinetic energy to potential energy, and the pressure builds up to boiler pressure before the end of the delivery tube is reached. The water then flows through a check valve to the boiler. When starting, steam and air are forced through the automatic ring valve R and check valve P to the atmosphere. As soon as water reaches the combining tube, the pressure drops, owing to condensation. The water then continues through the delivery tube to the boiler; a partial vacuum is formed in the annular space around tubes C and D, and valve P closes.

Injectors may be classified as *lifting* and *non-lifting*, depending on whether or not a partial vacuum is created in the suction pipe when starting. A non-lifting injector must always be placed below its source of feedwater supply because of its inability to create a partial vacuum. Injectors that have one set of nozzles for lifting the water and another

set for forcing it into the boiler are called double-tube injectors. Those

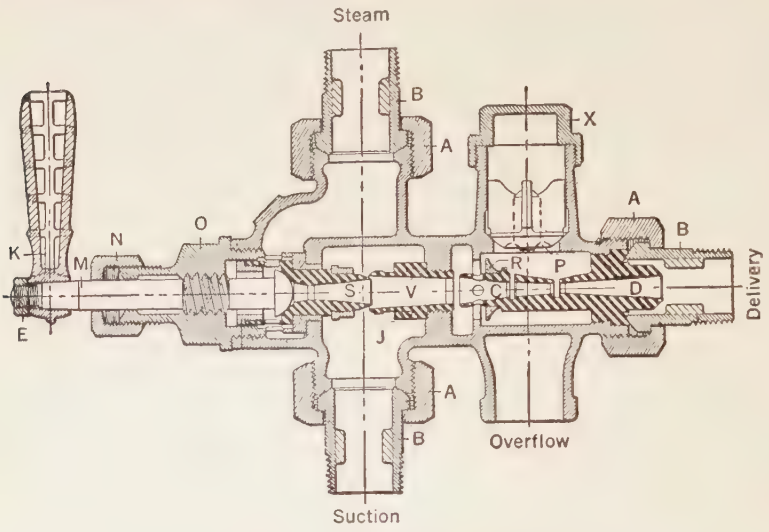


Fig. 93.—Section of a Metropolitan steam injector.

which accomplish the same result with one set of nozzles are called single-tube injectors, Figs. 93 and 94. A single-tube injector will lift water at 70 deg. F. about 20 ft. and water at 100 deg. F. about 14 ft. A double-tube injector will lift water at 120 deg. F. about 20 ft. and will deliver it against a boiler pressure of 350 lb. per sq. in.

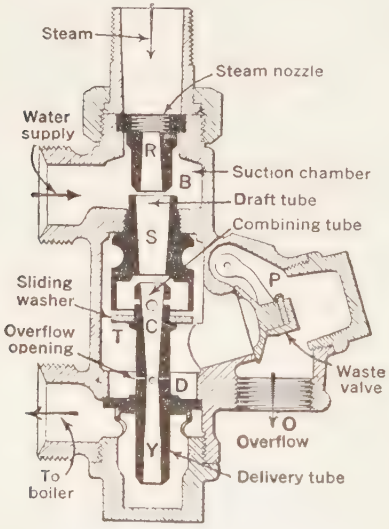


Fig. 94.—Penberthy single-tube automatic injector.

Though much energy is lost in friction and at impact of water and steam, the energy available in the expansion of one pound of steam is enough to force the condensed steam and many pounds of feedwater into the boiler. The injector is an efficient and convenient device, cheap and compact, with no moving parts, which delivers hot water to the boiler. In locomotives, injectors are frequently used for feeding the boiler, as they require very little space and provide a means of warming the feedwater, the exhaust steam from the engines being used to pro-

duce draft. In stationary practice the use of injectors is limited to small boilers or as a reserve with feed pumps. Some of the disadvantages of an injector are: inability to handle hot water, difficulty of maintaining a continuous flow under extreme load variation, and the unreliability of operation under certain conditions. From a purely thermodynamic standpoint, the efficiency of an injector is nearly 100 per cent, since the heat of the steam is returned to the boiler, less a small radiation loss. However, as a pump the injector requires more steam per pound of water handled than is required by inefficient plunger pumps.

PROBLEMS

1. Steam of 98.5 per cent quality at 350 lb. per sq. in. abs. pressure entered a radiant type superheater. If the steam left the superheater with a temperature of 700 deg. F., how much heat was added per pound in the superheater?

2. A steam generator delivered 28,000 lb. of superheated steam per hour at a pressure of 550 lb. per sq. in. abs. and a temperature of 800 deg. F. (*a*) Find the degrees of superheat. (*b*) If the steam left the boiler drum with 2 per cent moisture, how many boiler horsepower were developed in the superheater?

3. A boiler unit generated steam of 98 per cent quality at 200 lb. per sq. in. abs. pressure, from feedwater at a temperature of 210 deg. F., and superheated it 300 deg. F.; what percentage of the fuel burned was required to superheat the steam? Neglect all losses.

4. A generating unit evaporated 0.7 lb. of water per cu. ft. of natural gas. Feedwater temperature 185 deg. F. and steam pressure 180 lb. per sq. in. abs. The steam left the boiler and entered the superheater with a quality of 99 per cent; the steam left the superheater at 150 deg. above the vaporization temperature. (*a*) Find the heat absorbed by the boiler and (*b*) by the superheater in B.t.u. per cubic foot of gas. (*c*) What percentage of the total capacity of the unit was developed by the boiler, and (*d*) by the superheater?

5. Steam at a pressure of 125 lb. per sq. in. abs. and 96 per cent quality was passed through a superheater. Two and four-tenths pounds of flue gas passed over the tubes per pound of steam passing through the tubes. If the gases were cooled 250 deg. F., find the final temperature of the steam. Assume mean specific heat of gases as 0.25.

6. The average temperature of the products of combustion passing over a superheater was reduced from 1100 to 850 deg. F. If 12 lb. of flue gases were produced by each pound of fuel, how many pounds of steam were superheated from saturation to a final temperature of 600 deg. F. for each pound of fuel burned? Steam pressure, 165 lb. per sq. in. abs. Neglect all losses.

7. Coal contained 12,500 B.t.u. per lb., 8 lb. of steam were generated per pound of coal, and 15 lb. of gases were produced per pound of coal. If the superheat gained was 100 deg. F. at a pressure of 400 lb. per sq. in. abs. and the efficiency of heat transfer in the superheater was 85 per cent, find the reduction of temperature of furnace gases in passing across the superheater. Furnace gases reach the superheater at 1600 deg. F.; mean specific heat of gases, 0.25.

8. Boiler furnace gases were cooled from 1800 to 1500 deg. F. in passing over a superheater. The dry coal analysis was C, 80; H, 5; O, 7; N, 0.9; S, 1.1; ash, 6; and the excess air was 80 per cent. Carbon in the ash and refuse 2 per cent of the

weight of coal used. Calculate the degrees of superheat at 200 lb. per sq. in. abs., if steam of 98 per cent quality entered the superheater; 10 lb. of steam were superheated per pound of dry coal and 90 per cent of the heat lost by the furnace gases was absorbed by the steam. Mean specific heat of gases, 0.25.

9. How many square feet of surface are required with an interdeck superheater if it is to superheat 60,000 lb. of steam per hour under the conditions of problem 1? Each square foot of surface will transfer 1600 B.t.u. per hour from the hot gases to the steam.

10. If 600 B.t.u. per hour per sq. ft. of surface were transferred from the hot gases to air in a plate type air heater, find the number of square feet of surface required to heat 200,000 lb. of air per hour from 70 to 420 deg. F.

11. The temperature of the air supplied to a boiler furnace for combustion was increased from 80 to 350 deg. F. in passing through a tubular air heater. There were 180,000 lb. of air passing through the heater per hour and 188,000 lb. of flue gases per hour. (*a*) Determine the final temperature of the flue gases leaving the heater, if the initial temperature was 570 deg. F. Specific heat of air, 0.24; specific heat of gases, 0.25. (*b*) How much heat was removed from the flue gases per hour?

12. What would have been the final temperature of the flue gases of problem 11, if 10 per cent of the heat given up by the gases was lost by radiation to the external air?

13. The air supplied for combustion in a boiler furnace passed through an air heater. The ratio of the heat gained by the air to the heat lost by the flue gases was 0.90. Mean specific heat of the air and gases, 0.24 and 0.25, respectively. The coal used had the following ultimate analysis, per cent by weight: C, 79.44; H, 4.29; N, 1.56; O, 2.84; S, 1.11; M, 3.42; and A, 7.34. Flue-gas analysis, per cent by volume: CO_2, 11.6; O, 6.8; CO, 0.8; and N, 80.8. Neglect the carbon in the ash and refuse. The temperatures of the flue gases and air entering the heater were 550 and 70 deg. F., respectively. If the flue gases left the heater at 300 deg. F., what was the temperature of the air as it left the heater?

14. An injector of modern design delivered 16 lb. of water per lb. of dry steam under the following conditions: initial steam pressure, 115 lb. per sq. in. abs.; quality of steam, 96 per cent; water supply at 60 deg. F.; delivery temperature, 140 deg. F.; delivery pressure, 150 lb. per sq. in. abs.; suction lift, 3 ft. Find the heat required to deliver one pound of water to the boiler.

15. There were 13 lb. of water at 90 deg. F. entering an injector per lb. of steam supplied. The steam (dry) was supplied at a pressure of 150 lb. per sq. in. abs. What was the temperature of the discharge from the injector, neglecting radiation and friction?

CHAPTER IX

FEEDWATER TREATMENT AND FEEDWATER HEATERS

115. Feedwater Impurities.—Natural waters are more or less contaminated as falling raindrops absorb oxygen, carbon dioxide, nitrogen, dirt, and other impurities from the air. Water thus polluted is a solvent to a greater or less extent of the minerals of the earth. This condition is augmented by the addition to the water of acids formed by decaying organic materials. Furthermore, the waters of lakes and rivers are frequently contaminated with industrial wastes and sewage so that they carry acids, alkalies, oil, dirt, organic matter, and débris, in addition to the impurities collected by natural causes. In commercial power plants, waters of varying degrees of purity must be used. The need for water of greater purity is more urgent in large boiler units and in plants where boilers are driven at high steaming rates. Small and moderate-sized plants, operating without undue forcing and with intermittent service, may be able to use untreated water of good quality or poor water with a limited amount of treatment.

Feedwater impurities may be responsible for the following ill effects in boiler and power-plant operation: (1) reduction of the amount of heat transmitted to the boiler, due to scale formation on the boiler heating surfaces; (2) tube and plate failures caused by the reduced amount of heat transmitted through them; (3) embrittlement or corrosion of the boiler steel; (4) bad operation, foaming and priming; (5) cost of cleaning, repairs, inspection, maintenance, and reserve equipment; (6) heat losses due to frequent boiler blow-downs; (7) loss of economy of steam using equipment caused by dirty steam; and (8) additional cost of condensing equipment for gas-laden steam.

Most of the foregoing ills may be attributed to one or more of these items: scale, corrosion, embrittlement, foaming and priming, and occluded gases.

Scale is a deposit, on the internal surfaces of the boiler, of mineral materials and dirt. The nature of the scale varies from soft to very hard and flinty, and its effect, even with small deposits, is to insulate the metal surfaces and to reduce the boiler capacity. Such insulation may cause overheating and weakening of the boiler tubes and shell plates.

Corrosion produces a wasting away of the boiler parts with a reduction of their strength. The production of such a condition, resulting from the tendency of the iron of the boiler metal to go into solution, is accelerated by acidity of the boiler water, and the action is prolonged by the presence of oxygen.[1] Condensed steam from a boiler in which carbon dioxide is liberated is likely to have an acid reaction and to accelerate corrosion. In some cases, corroded boilers may be operated with reduced pressures, depending upon the amount of the decaying action; in others, corrosion may necessitate the condemnation of the boiler for steam-generating purposes.

Caustic embrittlement occurs in boilers operating with waters abnormally high in free sodium bicarbonates or waters carrying free sodium carbonates without sulphates. These materials hydrolyze under heat and pressure into sodium hydroxide and free carbon dioxide. The sodium hydroxide frequently concentrates in boiler seams and joints to such an extent that it attacks adjacent metal that may have been highly stressed during construction. Cracks in the boiler plates result from the combined stress and chemical attack. Thus far caustic embrittlement has occurred only in riveted boilers.

Foaming and priming may occur in combination. Suspended organic matter, some oils in the presence of soda salts, or waters containing concentrations of soluble sodium salts may cause the water in the boiler to fill the steam spaces with a mass of frothy bubbles. When this occurs the term foaming is applied. Priming is the discharge of slugs of water with the steam due to the causes of foaming, or it may occur in boilers having small steam and water disengaging surfaces. The rapid ebullition of water in such boilers causes water to be thrown up into the steam space and to be carried away with the steam. Both foaming and priming cause dirt and other objectionable material to be carried by the steam to foul superheaters, traps, steam lines, and turbine blading.

116. General Effects of Impurities. -Some of the common impurities in boiler feed waters and their effects are as follows: *Oxygen* induces pitting and corrosion of all metal parts with which it comes in contact. This is especially true at high temperatures. *Carbon dioxide* alone is likely to produce corrosion or to enter into combination with other materials to produce scale-forming compounds. *Gases* such as oxygen, nitrogen, carbon dioxide, and others not condensable at the usual condenser pressures and temperatures, which may be carried by the steam, impose an additional load upon the air ejectors or air pumps of

[1] "Some Fundamental Considerations of Corrosion in Steam and Condensate Lines," by P. E. Hall and A. R. Mumford; *Heating, Piping and Air Conditioning*, vol. III, No. 11 and 12.

condensers. *Dirt* and *mud*, unless washed out of boilers frequently, are likely to bake on the interior surfaces and produce scale. *Acids* produce corrosive action within the boiler, which results in the destruction of the boiler parts or else renders them unsafe for operation. *Alkalies* in excess induce bad operating conditions such as foaming and priming, and also under certain conditions may produce either corrosion or embrittlement of the boiler parts. *Oils* produce foaming and priming in a boiler and may coat the interior surfaces so that the heat transfer from the plates and tube walls to the boiler water is greatly retarded. Burning and weakening of the boiler parts results. *Fatty oils* break up into acid compounds, producing corrosion. *Dissolved organic matter* may break up into acids or into alkalies and thus produce very undesirable results. *Dissolved mineral materials* are usually precipitated by heat or left as a residue after the evaporation of the water and thus produce boiler scales of different degrees of hardness.

117. Water Hardness.—Hardness is a relative term applied to waters; it is indicative of the presence of scale-forming materials. The general classifications of hardness are *temporary* and *permanent*.

Waters containing only bicarbonate compounds of magnesium and calcium are classed as having temporary hardness. The bicarbonate compounds are formed by waters impregnated with carbon dioxide dissolving, as they seep through limestone, either or both magnesium carbonate, $MgCO_3$, or calcium carbonate, $CaCO_3$. The CO_2 in solution has the property of a weak acid, and the resulting bicarbonates $Mg(HCO_3)_2$ and $Ca(HCO_3)_2$ are produced. Water alone is only a slight solvent for either $MgCO_3$ or $CaCO_3$.

The application of heat to waters having temporary hardness will cause the CO_2 content of the bicarbonates to be driven off, with the result that the carbonates are precipitated. This process is nearly complete at a temperature of 212 deg. F., so that waters with only temporary hardness can be softened by heating. The resulting sludge caused by the precipitation may be blown out of the boiler or removed from the feedwater heater. The degree of softening is affected by the average temperature to which the waters are subjected. Heating to a temperature of 210 to 211 deg. F. in an open type of feedwater heater may be sufficient. The scales formed by untreated or insufficiently treated waters of temporary hardness are softer and more porous than those produced by permanently hard waters.

Permanent hardness is a term applied to waters having scale-forming materials not precipitated by heat alone at temperatures less than 300 deg. F. Permanently hard waters contain magnesium sulphate, $MgSO_4$, or calcium sulphate, $CaSO_4$, or both.

Boiler feedwaters may also contain chlorides and nitrates of calcium

and magnesium. Under heat and pressure the chlorides and nitrates are prone to break up to form corrosive acids. Sodium carbonate, Na_2CO_3, sometimes appears free in raw waters and quite often in waters overtreated with soda ash. Treatment by the zeolite process produces free sodium carbonate in some waters. When heated, sodium carbonate is very likely to be decomposed to a hydroxide or caustic soda, NaOH, and overstressed or strained boiler parts are embrittled by the hydroxide.

The relative hardness of waters is expressed in terms of equivalent grains of scale-forming material per U. S. standard gallon. The amount of material in untreated waters may be from 2 to 140 grains per gallon. Waters are classified, according to their content of scale-forming materials per gallon, as follows: less than 8 grains, very good; 8–15, good; 15–20, fair; 20–30, poor; and more than 30, bad. A complete indication of the desirability of a water for boiler-feed purposes is not given by the content per gallon of the equivalent grains of scale-forming materials, as such materials are not all of equal undesirability. Furthermore, some waters either untreated, partially treated, or overtreated, may carry the ions of inoffensive salts which are likely to produce destructive combinations within the boiler.

118. Objects of Water Treatment. —The primary objects of water treatment are twofold: (1) the removal of suspended and soluble solids and (2) the removal of gases. For the best results the purification of the water should take place outside of the boiler. Frequent blowing down of a boiler to remove concentrated solutions of salts and precipitated sludges results in a loss of heat in the water and materials discharged. Such losses may be lessened by the use of continuous blow-off devices which permit the incoming feedwater to be heated by the water and material discharged so that a part of the heat may be recovered.

Water treatment should reduce to a minimum or absolutely prevent the following undesirable and dangerous conditions: foaming and priming, scale formation, corrosion, and caustic embrittlement. In order to prevent corrosion the water should be free of oxygen, carbon dioxide, free acids, and materials which may break down to form acids. Care must be taken to prevent excessive concentrations in the boiler of hydrogen ions (H) and hydroxyl ions (OH). The former produce acid reactions and the latter alkaline reactions.

119. *p*H Values.—Measurement of water acidity or alkalinity is expressed by the *p*H value, which is the logarithm of the reciprocal of the hydrogen (H^+) ion concentration in grams per liter of water. Pure water ionizes to the extent of 10^{-7} gram of hydrogen (H^+) ions per liter, and as the water is neutral the hydroxyl (OH^-) ions will also amount to 10^{-7} gram per liter. The sum of the hydrogen- and hydroxyl-ion

concentrations should always equal 10^{-14}. Therefore, pH values of less than 7 indicate excess hydrogen (H^+) ions or acidity, and pH values greater than 7 signify an excess of hydroxyl (OH^-) ions or alkalinity.

The presence of hydroxyl ions to the extent of a pH value of approximately 9.6 is known to limit corrosion; therefore, boiler waters are often maintained definitely alkaline for this purpose.

Acid concentrations are reduced by the use of alkaline reagents such as barium, sodium, and calcium hydroxides. Excessive alkalinity is likely to produce foaming and priming and, in some cases, caustic embrittlement. To prevent the latter, definite ratios of sulphates to caustic material, such as sodium carbonate and its equivalent of sodium hydrate, should be maintained. The recommendations of the A.S.M.E. in regard to the ratios are dependent upon the boiler pressure and are 0–150 lb. sq. in., 1 to 1; 150–250 lb. sq. in., 2 to 1; 250 lb. sq. in. and over, 3 to 1. Caustic embrittlement of boiler steel may also be inhibited by the maintenance in the boiler water of the proper amounts of soluble phosphates, tannates, chromates, nitrates, acetates, etc.

120. Methods of Water Treatment.—Water treatment should be undertaken only after analyses of the water and recommendations for its treatment have been made by an experienced and competent chemist. There is no single prescribed process of treatment or boiler compound that is suitable for all waters. Each case must be considered individually.

Water-treatment processes include removal of débris by stationary or moving screens, filtration, separation of mud and silt in settling basins by subsidence, heating, evaporation or distillation, deaeration, treatment with lime, treatment with soda ash, treatment with both lime and soda ash, lime and barium, coagulants, and zeolite softeners. The processes are not all carried out in the same manner; some of the methods are continuous; others are intermittent. Cold water is treated in some cases, while heat is necessary for best results in many methods. The space available for the treating plant may be a deciding factor in the selection of the treating system. The method chosen depends upon the degree of purification necessary as determined from operating conditions, and the economic balance between the cost of treatment and the ultimate return on the investment necessary.

121. Filtration.—Suspended matter in water is often removed by the use of mechanical filters. The water is usually passed downward through layers of finely divided filtering media such as sand, quartz, or gravel. The introduction of coagulants into the water as it passes into the filter aids materially in the maintenance of a rapid rate of filtration. The common coagulants used are alum, sulphate of aluminum, or iron

sulphate. These chemicals when added to the water react with the alkaline salts produced by bicarbonates or carbonates of lime and magnesia to form a bulky gelatinous precipitate of hydrate of aluminum or iron. This gelatinous hydrate forms a blanket in the filtering material which effectively removes small particles of suspended matter. Automatic coagulant feeding devices are provided for the introduction of the coagulant into the water to be filtered.

The pressure filter is commonly used and may be of either the vertical or horizontal type. The units may be single or in multiple. A Graver vertical pressure filter is shown by Fig. 95. As is common with all pressure filters, this device consists of a closed cylindrical shell containing the filtering material. The water enters the filter body at the top and passes downward through the filtering media and the perforated strainer plate which supports the layers of the filter bed. The filtered water is removed from the bottom of the shell below the strainer plate. Suitable piping connections are provided so that the filter may be backwashed by introducing water under pressure below the strainer plate and forcing it upward through the filter bed to remove entrapped débris.

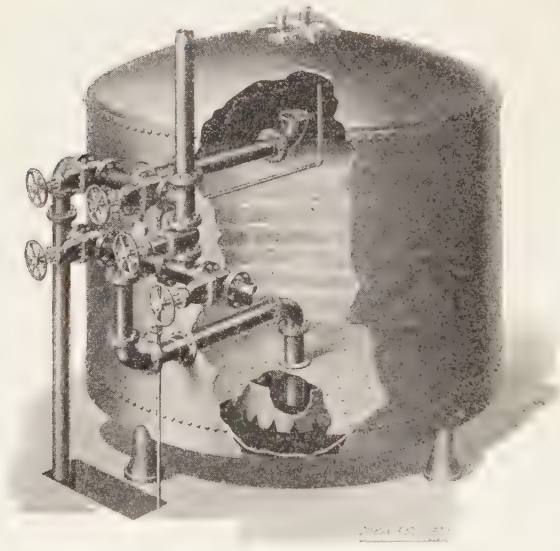


Fig. 95.—Graver vertical pressure filter.

122. Boiler Compounds. When composed of the proper ingredients, boiler compounds may prove beneficial. However, the ingredients of the compound must be adapted to the water used, as a single formula is not applicable to all waters. Many worthless or even dangerous products are marketed under the guise of boiler compounds. In any event, the use of compounds necessitates the removal of the solids from the boiler. Such palliatives do not remove occluded gases from the water.

123. Lime Treatment. Lime in the form of limewater, $Ca(OH)_2$, is suitable for the treatment of temporary or bicarbonate hardness. The carbonic acid present in the water either in a free state or in bicarbonate

combination is taken up and the insoluble precipitate of lime carbonate formed. The process is best carried out in large tanks from which the softened water may be decanted and the sludge removed from the treating plant.

124. Soda Treatment.—The soda process is suitable for waters containing sulphate hardness. Either caustic soda, NaOH, or sodium carbonate, Na_2CO_3, or both may be used. The soda ash, Na_2CO_3, causes the decomposition of the sulphates and the precipitation of insoluble lime carbonates and soluble magnesium carbonates. The addition of lime converts the soluble magnesium carbonate into an insoluble hydroxide. Caustic soda takes up free carbon dioxide and converts magnesium sulphate into insoluble magnesium hydroxide. Tri-sodium phosphate may be used as a reagent for water treatment in place of Na_2CO_3. Tri-sodium phosphate is more expensive than soda ash, but the water thus treated has a lower content of caustic material. The latter condition is particularly advantageous in high-pressure boilers.

125. Lime and Soda Process.—This is a common method employed where both bicarbonate and sulphate hardness exists. The lime added takes up the CO_2 not affected by the soda used in the treatment of the sulphates. Figure 96 illustrates a Vater hot-process treating plant which is suitable for the use of lime, or soda, or both. The lime and soda reagents are taken from the chemical tanks, and the properly proportioned amounts of each are introduced into the raw water as it enters the softener proper. The incoming water mingles with exhaust steam as it enters at the top of the softener and is heated to about 210 deg. F. The water slowly passes downward, and the precipitated material is deposited at the bottom of the softener shell where it may be removed through the sludge outlet. The treated water is withdrawn above the level at which the sludge collects and is normally passed through a filter before it goes to the boiler feed-pump suction. The use of sodium aluminate facilitates the action of the precipitating reagents and also serves as a coagulant in any lime or soda process of treatment.

126. Hot and Cold Processes.—Either heated or unheated water may be treated with lime or soda as reagents. In general, heating hastens the process of precipitation and requires smaller tank equipment. Owing to the lower solubility of calcium carbonate in hot water, the hot process usually gives softer water than is obtained by use of the cold process. Less lime is necessary with heated water as the bicarbonates are broken down by heat and the lime is used to precipitate the magnesium carbonates as a hydroxide. Where steam is not available for heating water the cold process may be employed with larger treating tanks and at a slower rate.

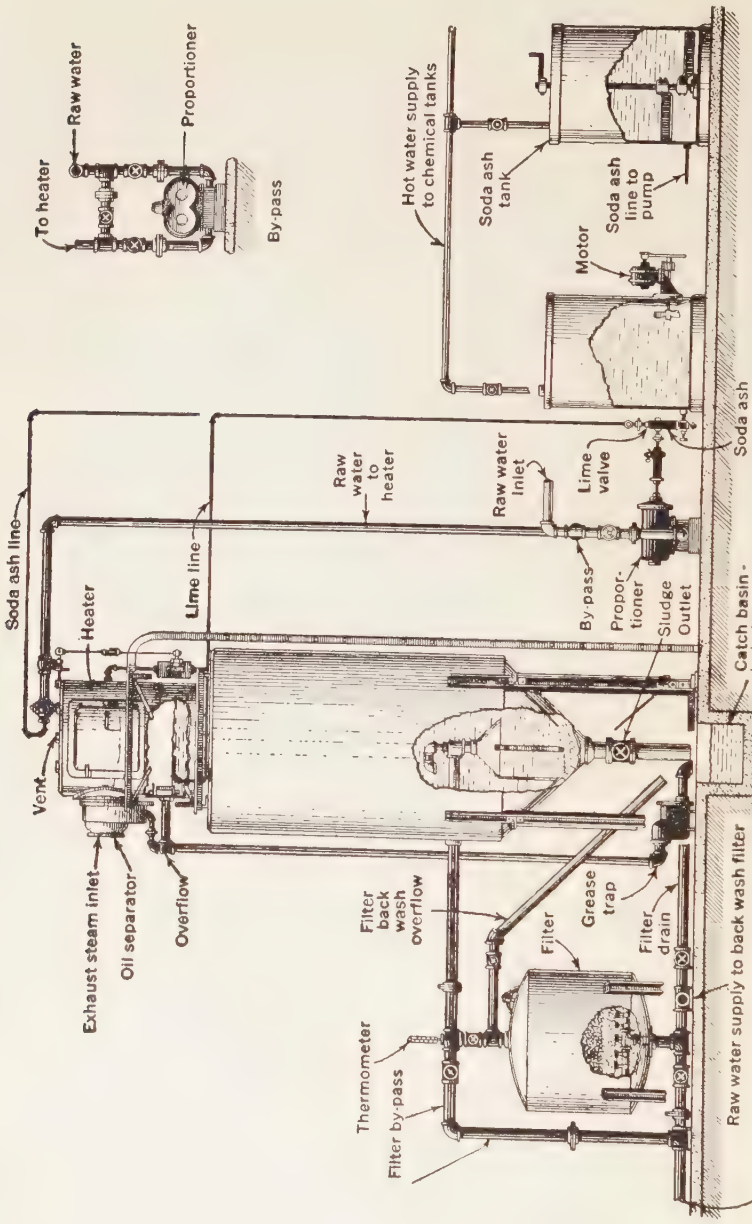


FIG. 96.—Water hot-process water-softening system.

127. Zeolite Treatment.—Natural or artificial hydrated silicates of aluminum having an alkaline base have the property of removing calcium or magnesium from water passed through them. These materials have the chemical composition of $2SiO_2 \cdot Al_2O_3 \cdot Na_2O$ and are known as zeolites. Their bases are interchangeable so that the sodium of the zeolite goes into solution as sodium carbonate or sulphate as the calcium and magnesium are absorbed from the water. No precipitates or sludges are formed in this process of treatment. The zeolite material is regenerated when it becomes inactive by washing it with salt (NaCl) brine. Waters for zeolite treatment must be cool and free

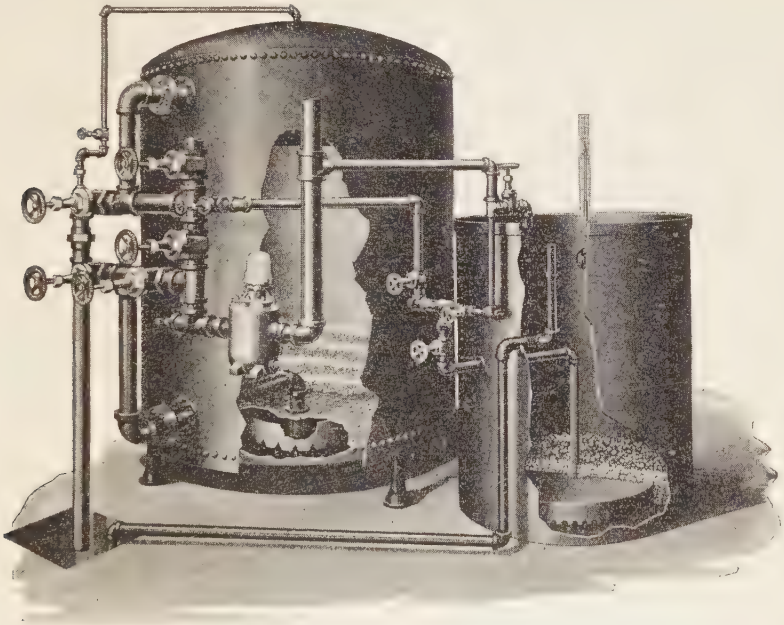


Fig. 97.—Graver vertical zeolite water-softener.

from débris, mud, silt, or iron, as these materials clog or coat the particles of zeolite. Where the water is dirty, acid, or where iron exists, the water must have prior treatment and purification. A sectional view of a Graver vertical zeolite softener is illustrated by Fig. 97. The softening equipment consists of two tanks as shown; the large shell with the dished heads is the softener proper; the water to be softened enters at the bottom of the shell and passes upward through the strainer plate. The strainer plate serves to distribute the water and supports the minerals held within the softener. The layers of materials from the strainer plate upward are: (1) coarse gravel, (2) medium-sized gravel, (3) fine gravel, (4) a thicker layer of quartz, (5) the zeolite bed. The flow of

water is upward through the filter beds and the zeolite materials. The softened water is taken from near the top of the tank at a distance above the zeolite bed to prevent loss of the zeolite materials. The open tank at the right of the figure is the brine tank and is used when the softener is reconditioned. The proper control valves for normal operation and for regenerating conditions are shown on the front of the apparatus.

As the process is an exchange, zeolite-treated waters carry a considerable quantity of sodium salts in solution. Waters thus impregnated are likely to cause foaming and priming in the boiler, and they may also cause caustic embrittlement of the boiler plates.

Certain waters should be first treated with lime to remove the carbonates and then put through the zeolite material to remove the permanent hardness or sulphate. The lime-zeolite process removes the danger of caustic embrittlement. Zeolite and untreated waters containing free sodium carbonate, Na_2CO_3, may be treated with sulphuric acid to increase their sulphate contents. The water should always be left slightly alkaline, and the treatment must be under the direction of someone capable of handling such work; otherwise, adding acid to the feedwater may be very dangerous.

128. Evaporators. Nearly pure water can be obtained by distillation of the feedwater supply. In plants where surface condensers are used and their condensate returned to the boilers, the distilled water to be provided is only that necessary to make up the quantities of water disappearing by leakage and unavoidable losses. Distillation of the make-up water supply only is commercially feasible. The use of distilled water almost eliminates the formation of scale and other feedwater difficulties in boiler operation. Even where distilled water is provided, small amounts of scale-forming materials may be carried into the boilers by condenser leakages and by impurities thrown over in the distilling equipment. There is no satisfactory commercial method of eliminating chlorides from water other than by distillation.

The distilled water necessary is obtained by the use of evaporators. Various types of this equipment are built to operate using steam as a heating medium either at high, moderate, or low pressures. The vapor produced by the evaporator may be condensed in the feedwater heaters by the incoming feedwater or in separate evaporator condensers using the feedwater to be heated as the cooling water.

Evaporators may accomplish the vaporization by any one of three methods, depending upon their construction. These types and methods are: (1) *flash type*, hot water is injected into a chamber under vacuum where vapor is produced; (2) *film type*, water in a thin film is passed over steam-filled tubes; and (3) *submerged type*, steam-filled tubes are submerged in the water to be evaporated.

Evaporators are also further designated as being single- or multiple-effect units, depending upon the number of evaporating stages through which the water is passed. Thus a single-effect unit is one in which the vaporization is completed in a single coil or stage. A double effect is secured by condensing the vapor from a single-effect unit in the coils of a second evaporator. The second evaporator operates at a lower temperature, and from it the vapor passes to the feedwater heater or the evaporator condenser. Multiple-effect evaporators utilize more of the heat of the steam supplied to them but they do not give any greater

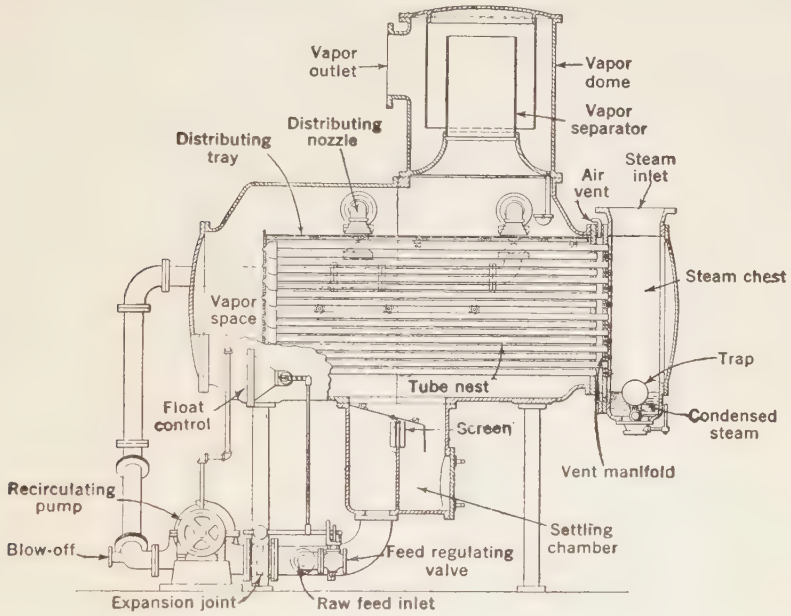


FIG. 98.—Foster Wheeler film-type evaporator.

evaporating capacity per square foot of heating surface than single-effect evaporators.

The low-pressure evaporators usually exhaust to open types of feedwater heaters at about atmospheric pressure; the high-pressure types of evaporators exhaust to special evaporator condensers.

A Wheeler film evaporator is illustrated by Fig. 98. The steam pressures maintained in the nests of return bend connected tubes may be at a pressure greater or less than atmospheric. In either case the condensed steam and the non-condensable gases of the steam are removed from the vent manifold. The water to be evaporated is sprayed continuously over the tubes by the action of a centrifugal recirculating pump forcing the water through nozzles above a perforated

distributing tray. The water level in the shell of the evaporator is always held below that of the lower tubes. High rates of heat transmission are obtained with small temperature differentials. The washing action of the water over the tubes tends to prevent the tubes from becoming scale coated. Scale is also removed from the tubes by reversal of the vapor and water flow, causing unequal expansion rates of the tubes and the scale. As a result, the scale falls from the tubes to the settling chamber. The vapor produced in the evaporator is carried away to a suitable evaporator condenser or heater.

129. Deaerators. Air, oxygen, carbon dioxide, or other occluded gases carried by water may be active in producing corrosion within

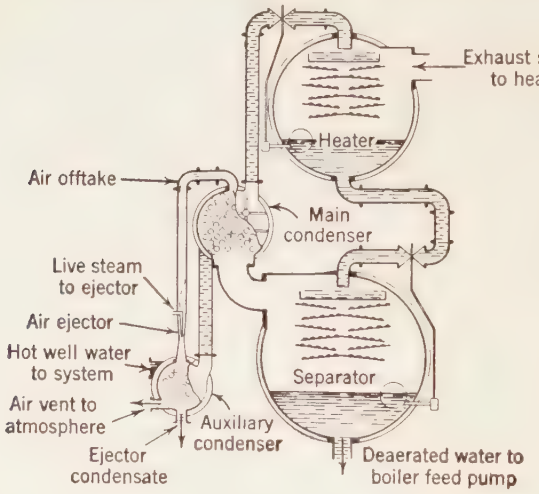


Fig. 99.—Elliott type "E" deaerator.

boilers, piping, heaters, and economizers. Mechanical devices used to liberate these gases are known as deaerators. In any event, the water is boiled within the deaerator, and in the process, air, carbon dioxide, and other gases are separated from the vapor as it condenses. Steam is used as the heating medium in deaerators at pressures ranging from high to less than atmospheric.

A feedwater heater of the open type inherently serves as a deaerator provided that the water is heated to a temperature high enough to drive off the gases held within the water. The gases escape through the vent of the heater. Several manufacturers equip either the open or the closed types of feedwater heaters with deaerating attachments, thus making a single unit serve as both a heater and a deaerator.

Deaerators are made in several styles depending upon the nature of the operating conditions of the plant. Figure 99 shows a diagrammatic

sketch of the cycle of operation of a Type E Elliott deaerator. Water from the main supply or hot well is passed through the auxiliary and main condensers of the deaerator, as it flows to the heater. Exhaust steam supplied to the heater produces the desired water temperature. From the heater the water is passed to a separator which is under a pressure less than atmospheric. The vacuum is always great enough so that the temperature corresponding to the absolute pressure in the separator is less than that of the water. As a result of the reduced pressure the water in the separator is boiled violently, and some vapor, together with the dissolved gases, is thrown off. The water cascades over the separator trays and is removed from the bottom of the chamber. The vapor and emitted gases pass to the main condenser where the vapor is reduced to water and the non-condensable gases are removed by a steam-actuated air ejector. The heat of the vapor and of the air ejector exhaust is recovered by the incoming water as it passes through the main and the auxiliary condensers.

130. Feedwater Heating.—The economical production of power necessitates the utilization of a maximum amount of the heat that may be obtained from a fuel. In steam-boiler operation the flue gases always leave the last pass of the boiler setting at a temperature well above that of the saturated steam and the water within the boiler. In many but not all plants a part of the heat of the escaping flue gases may be profitably employed to heat the boiler feedwater or to preheat the combustion air.

A considerable quantity of exhaust steam, with a large heat content, is available in plants having the prime movers operating non-condensing. Steam-engine- or turbine-driven auxiliaries such as draft fans, boiler feed pumps, electrical exciter units, air compressors, etc., provide additional exhaust steam for heating purposes. A part or all of such exhaust steam may be profitably used for heating water.

Some large central stations operate with all the auxiliaries electrically driven by a house turbo-generator exhausting into a condenser using feedwater for cooling water. The exhaust supply may be inadequate so that high-pressure steam or steam bled from the various turbine stages is used with economical results for feedwater heating.

The distinct advantages that may accrue from feedwater heating are enumerated by the following items: (1) reduction of boiler strains; (2) utilization of heat otherwise likely to be lost; (3) partial purification of untreated feedwater; (4) closer approximation of ideal thermal efficiencies for the plant by bleeding steam from turbine stages; (5) relief of the last stages of steam turbines from great volumes of low-pressure steam by stage bleeding to feedwater heaters; and (6) increased boiler capacity.

131. Economy of Feedwater Heating.—Many boilers do not show an increase of efficiency when supplied with heated water; others show an efficiency increase or decrease due to the change in the rate of heat transfer. In any event, the saving due to feedwater heating appears in the reduced consumption of fuel and the increase in overall efficiency for the plant. The amount of the fuel saving is dependent upon the initial and final temperatures of the feedwater, and the total heat content of the steam produced, where waste heat is used for water heating. The saving effected is a variable dependent upon the operating conditions of the plant.

The fuel saving due to feedwater heating may be expressed by:

$$S = \frac{(h_{f,2} - h_{f,1})\,100}{(h_g - h_{f,1})}, \tag{90}$$

where S = fuel saving, per cent.
h_g = total heat of the boiler steam, B.t.u. per lb.
$h_{f,1}$ = initial heat of liquid of the feedwater, B.t.u. per lb.
$h_{f,2}$ = final heat of liquid of the feedwater, B.t.u. per lb.

132. Feedwater Heater Classifications.—Divisional groupings of feedwater heaters may be made with reference to: (1) heat supply; (2) method of heat utilization; (3) steam pressure; and (4) steam-flow arrangements.

When the heat supply is from hot flue or waste gases, a water heater is designated as an *economizer*. When the heat utilized is derived from exhaust, bleeder, gland, jet, or live-steam sources the term *feedwater heater* is applied to the equipment. Feedwater heaters of the *open type* utilize the heat of the steam by the direct contact of the steam and water as the mixture is made. Open heaters operate at pressures ranging from atmospheric to 30 lb. per sq. in. gage. The steam and the water are at the same pressure.

In the *closed type* of heater the heat of the steam is transmitted through metal walls. The steam and water are not in direct contact. Closed heaters may operate at any steam pressure ranging from less than atmospheric to high. In closed heaters the water and steam pressures are not usually the same.

Heaters operating at pressures less than atmospheric are termed *vacuum* or *primary*. When the steam pressure is atmospheric or only slightly above atmospheric, heaters are classed as *atmospheric* or *secondary*. The *pressure* type of heater, as the name implies, may carry any reasonable pressure above that of the atmosphere.

When the steam is drawn into the heater by the partial vacuum produced by steam condensation, the heater is of the *induced type*.

Steam is taken into the heater only as it is required. In the *through* type of heater the steam is forced through the heater irrespective of the amount of steam condensed or the requirements for steam.

133. Open Feedwater Heaters.—These heaters may be built as either horizontal or vertical units. Either type requires the following principal parts: an enclosing cast-iron or steel shell, a steam and oil separator at the steam inlet, water trays or pans over which the feedwater overflows or is sprayed, steam and water controls, oil trap, excess-steam outlet, boiler feed pump connection, blow-off connections, and air and steam vent. A small amount of water-storage space is always provided

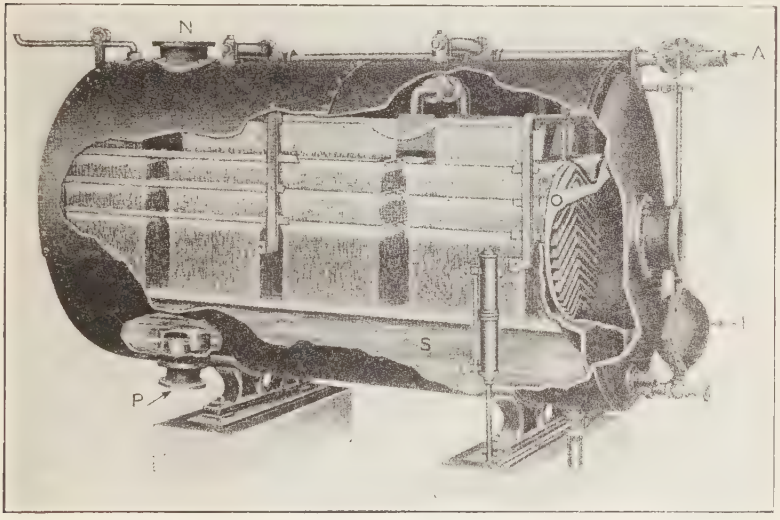


Fig. 100.—Hoppes horizontal open-type feedwater heater.

at the bottom of the heater, and in some cases a water filter is located within the heater shell.

A Hoppes horizontal open type of heater is illustrated by Fig. 100. The water enters the heater from the line *A*, and the amount of water entering the heater is controlled by a float located in the box *F*. The water is supplied to the uppermost cast-iron pan from which it overflows, falling successively into the pans beneath and finally to the storage space *S*. Steam which has passed over the oil separator *O* at the inlet fills the heater and surrounds the trays. The heat of the steam is given up to the flowing water. The trays can be taken out for the removal of precipitated scale deposits. Any excess steam supplied and gases liberated escape at the outlet *N*. The outlet *P* is placed in the bottom of the steel shell. This heater serves to deaerate the water, however,

the effectiveness of the degasification process is dependent upon the water temperatures attained.

The Hoppes open heater shown by Fig. 101 is similar to that illustrated by Fig. 100. This heater includes more positive deaerating equipment. A steam coil is located in the water-storage space. Steam, at 5 lb. per sq. in. gage or more pressure, is supplied to the coil, and the water within the heater is made to boil. The vapor produced, together with the gases liberated, is carried to a condenser *C* cooled by the incoming water. The non-condensable gases escape through a vent from the condenser, and the condensed vapor is returned to the heater.

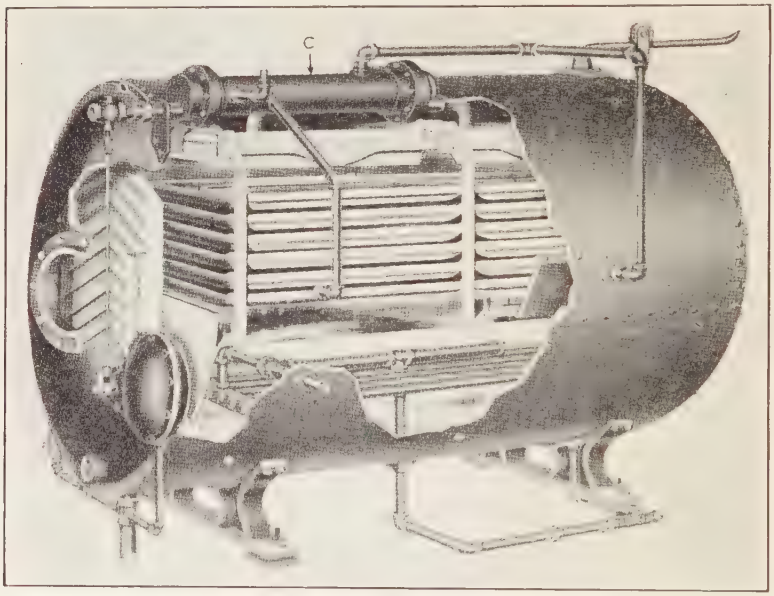


Fig. 101.—Hoppes horizontal deaerating feedwater heater.

A Cochrane vertical open-type heater with cast-iron shell is shown by Fig. 102. The incoming steam passes over a fluted oil separator and fills the heater body. The water to be heated enters at the top of the shell and falls as a spray into the steam which fills the heater. As the water passes downward it is directed over inclined and staggered cast-iron trays. The trays are both serrated and perforated, thus providing surface for breaking up and heating the water as well as for the liberation of gases. The trays are removable so that scale deposits may be taken out. The bottom of the heater serves as a storage tank, and the amount of water admitted to the shell is under the control of a float-actuated valve in the supply line. The heater shown has piping connections for condensation returns either by pump or gravity. Filtering

material is provided in the water-storage space. When the heater is operated under steam pressures greater than atmospheric the vent pipe should be connected to a condenser. The use of the condenser will permit the escape of liberated gases and allow the return of water which otherwise would escape as steam. Floating débris is discharged through a trap which also eliminates the oil removed by the oil separator. Cochrane heaters are also built with special deaerating attachments.

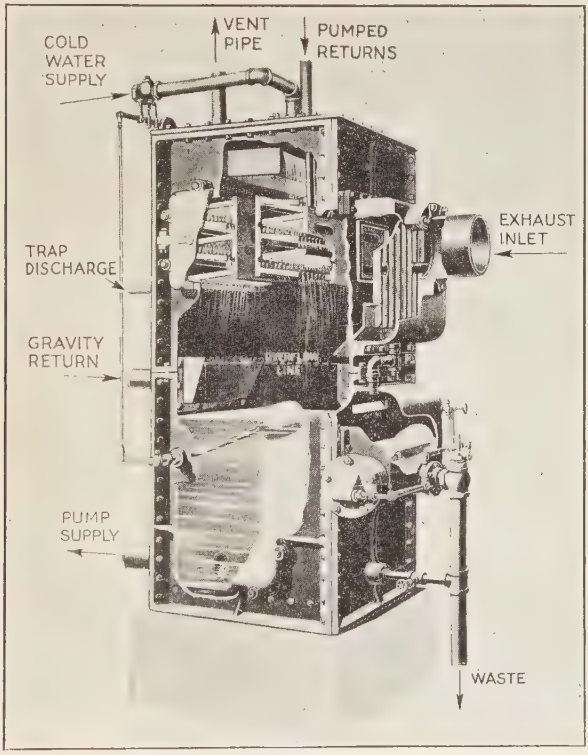


Fig. 102.—Cochrane vertical open-type feedwater heater.

All heaters of the open type must be located well above the pump suction so that water from them will flow freely to the pump by gravity, as pumps will not lift hot water by suction. Open-type heaters may be operated with considerable steam pressure. In any event, either the heater or the steam supply line should be provided with relief connections so that excessive and dangerous steam pressures will not be applied to the heater shell.

134. Jet Heaters.—These heaters employ more or less the principle of the jet condenser. As the steam and the water to be heated are mixed together, they are considered under the scope of open heaters.

An interior view of a Cochrane jet heater with two sets of spray nozzles C is shown by Fig. 103. The water inlets of this heater are located at the points marked A. Figure 104 presents the exterior view of a Cochrane jet heater with duplex water belts. The water inlets are designated by A and the steam inlet by B.

The water to be heated is sprayed through the nozzles into a chamber filled with steam. The steam may be under pressure or a vacuum. The pressure of the steam may be rapidly changed from one condition

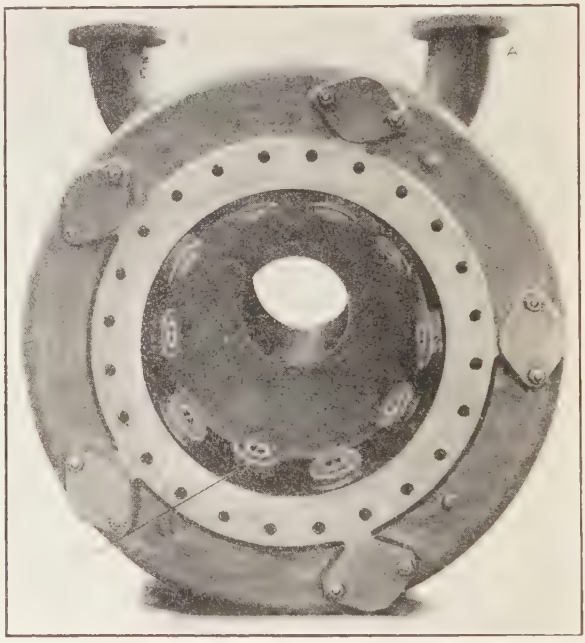


FIG. 103.—Cochrane jet heater, interior view.

to the other with variations in the flow of the feedwater or the supply of the exhaust or bleeder steam.

The water is thoroughly heated as it falls in small drops through the steam and passes to the hot-water storage reservoir or hot well. These heaters have no particular advantages over the ordinary open types of heaters in sizes of less than 5000-hp. capacity. Adjustments of the water-spray nozzles must be made as the demand for water increases or decreases.

135. Water Temperatures of Open Heaters.—Theoretically the heat given up by the steam should equal the heat absorbed by the water.

$$W_s(h_g - h_{f,2}) = W_w(h_{f,2} - h_{f,1}), \tag{91}$$

where W_s = weight of steam condensed, lb.
h_g = total heat of the steam as condensed, B.t.u. per lb.
$h_{f,1}$ = heat of liquid of inlet water, B.t.u. per lb.
$h_{f,2}$ = heat of the liquid of outlet water, B.t.u. per lb.
W_w = weight of water heated, lb.

Expressed in terms of temperatures, assuming that the specific heat of the water is 1, equation 91 becomes

$$W_s(h_g - t_2 + 32) = W_w(t_2 - t_1), \tag{92}$$

but t_2 can never exceed the steam temperature. t_1 and t_2 equal the initial and final water temperatures, deg. F. In actual operation there are certain inevitable heater losses. Consequently the value of W_s will always be greater than the amount calculated from the equation for a given quantity of water raised from a temperature of t_1 to t_2, by steam of a given pressure and quality. The heater losses may range from 5 to 10 per cent of the heat content of the steam actually used in the heater.

The weight of water leaving the heater is the sum of the weight of water supplied and the weight of steam condensed in the water.

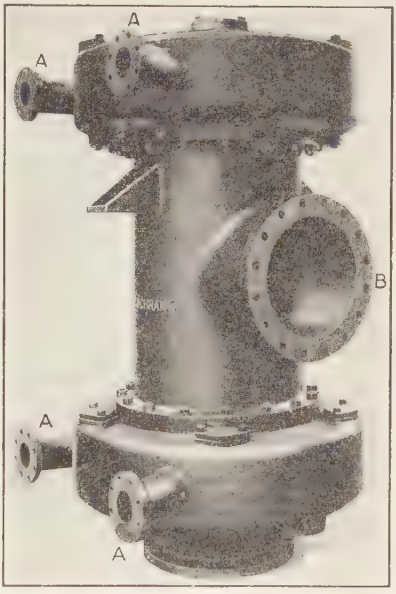


FIG. 104.—Cochrane jet heater, exterior view.

136. Closed Feedwater Heaters.—The steam and water are never in contact in the closed type of heater. Consequently they may be operated at pressures ranging from high to very low. The essential parts of a closed heater are a shell and the enclosed tubes or coils through which either steam or water is circulated. Heaters having the water inside the tubes or coils are called water-tube heaters; if the steam is confined within the tubes the designation of steam-tube heater applies. The water-tube heaters are those most commonly used: the steam and water flow may be in the same or in opposite directions in either heater, giving either parallel or counterflow conditions. In single-flow heaters the water flows in one direction only; multiflow heaters have the water flowing back and forth in various nests of the tubes. Closed heaters are ordinarily placed in the line between the boiler feed pump and the boilers.

Figure 105 illustrates a Schutte and Koerting closed heater that may be installed in either a vertical or horizontal position. This heater is of the multiflow water-tube type. The straight water tubes are packed into tube sheets at the top and bottom of the enclosing steel shell. The lower tube sheet is not attached to the shell, so that provision for expansion and contraction of the tubes is made.

Water enters the top header which is divided into compartments. The feedwater flows into one compartment and passes downward through one-fourth of the tubes to a compartment of the bottom header

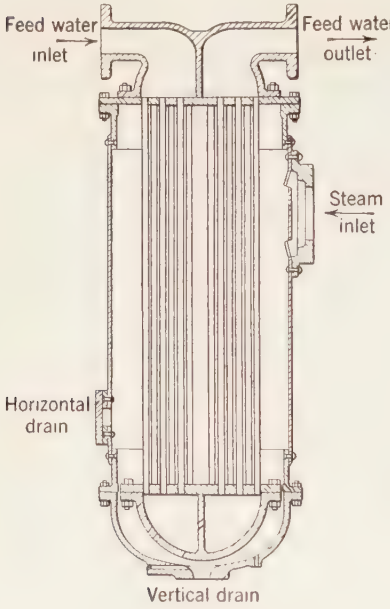


Fig. 105.—Schutte and Koerting closed-type feedwater heater.

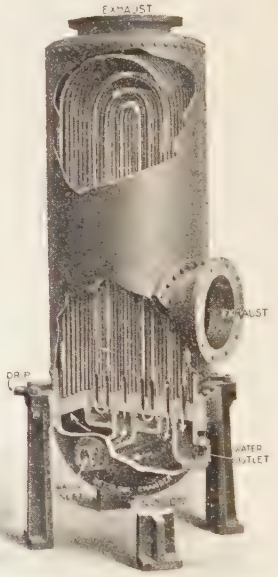


Fig. 106.—Patterson-Berryman feedwater heater.

fastened to the lower tube sheet. The water makes two passes downward and two upward through the four-tube nests before leaving the heater. Heating steam enters the top of the shell and surrounds the tube nests. The condensation is removed through a drain connection at the bottom of the shell. The assembled tube nests may be removed for inspection and cleaning without disconnecting the steam piping.

A Patterson-Berryman bent-tube type of closed feedwater heater is shown by Fig. 106. This heater consists of a cast-iron or steel shell containing a nest of inverted U-shaped circular copper tubes. The bent-tube construction takes care of strains produced by expansion and contraction of the small water passages. The heating surface is divided

into two to seven rows of tubes, depending upon the capacity of the heater. The water in the tubes passes through the surrounding steam as many times as there are rows of tubes. The small, smooth copper tubes permit a rapid transfer of the heat of the steam to the water. The lower part of the heater serves as a settling chamber, from which precipitated impurities may be removed.

137. Heat Transfer in Closed Heaters.—The principles of heat transmission in closed feedwater heaters and surface condensers are essentially alike. If liberal factors are allowed in the design of closed heaters the laws may be taken as the same as those for surface condensers, see Art. 267.

The amount of heat given up in the heater per square foot of heating surface is dependent upon the velocity of the water through the heater and the mean temperature difference between the steam and the water. An increase in either or both of these items will increase the amount of heat taken up by the water. Sufficient time must also be allowed in order that the temperature rise may be a maximum under the conditions of operation. In general, all other conditions being the same, a heater having a large number of passages of small cross-sectional area will transmit more heat than one having a few passages of large cross-sectional area.

The amount of heating surface necessary to heat a given quantity of water through a desired temperature rise is expressed as

$$A = \frac{W_w c_{pw}}{U} \log_e \left(\frac{t_s - t_1}{t_s - t_2} \right), \tag{93}$$

where A = total heating surface, sq. ft.
c_{pw} = specific heat of the fluid heated.
U = coefficient of heat transmission, B.t.u. per sq. ft. of surface per hour, per deg. F.
t_1 = initial temperature of the feedwater, deg. F.
t_2 = final temperature of the feedwater, deg. F.
t_s = temperature of the steam supplied, deg. F.
W_w = weight of water heated per hr., lb.

The values of U may range from 120 to 1200 B.t.u. per sq. ft. per hour per one degree Fahrenheit temperature difference. Average values of U depending upon the heater type, the steam and water temperature differences, and the velocity of the water flow, range from 200 to 600 B.t.u.

138. Economizers.—Assemblages of water-filled pipes placed in the path of flue gases, after the gases have left the boiler heating surfaces, are used to heat feedwater and are known as *fuel economizers.*

Such devices may be between the boiler setting and the chimney or they may be incorporated within the boiler setting as an integral part of the boiler. Under certain conditions integral economizers are spoken of as steaming economizers. The use of water heaters of this type is justified only in plants large enough to show a profitable saving after the interest, depreciation, and maintenance charges for the economizer are paid. Economizers increase the resistance offered to the flow of flue gases and also reduce the flue-gas temperatures. Consequently, the natural draft provided by a chimney is often inadequate and induced draft must be maintained. The additional costs incurred through the use of induced draft must also be given attention when the possible savings resulting from the use of economizers are to be considered.

Two classes of economizers are in use, those having cast-iron tubes and those having steel tubes. Cast-iron tube economizers may be used with boilers operating at 300 lb. per sq. in. pressure or less. The use of higher steam pressures necessitates the use of steel tubes. In the earlier days of the economizer, cast iron was used exclusively as the best non-corrosive tube material. Oxygen carried by the water passing through steel tubes induces rapid corrosion of the steel. Also the sulphur dioxide, SO_2, of flue gases is changed to H_2SO_4, a corrosive acid, when the flue-gas temperatures are reduced so that their steam content is condensed to water. The acid formed induces rapid corrosion on the exterior of steel or wrought-iron tubes. Modern steel economizer tubes may be protected against corrosion within and without by protective coatings, or externally by cast-iron coverings, also by deaeration of the water as a relief from internal corrosion.

139. Cast-iron Economizers.—Figure 107 illustrates the side elevation of a cast-iron tube economizer in connection with a water-tube boiler. Vertical cast-iron tubes are joined into headers or manifolds at the tops and bottoms. Such an assembly forms a section which may be had in various heights and widths. The sections when joined together by connections at both the top and the bottom may be erected to form an economizer of any feasible length. The tube arrangement in the interconnected sections may be staggered or the tubes may be directly in front of each other.

The water to be heated is introduced into the wall box or lower section connection at the flue-gas exit end of the economizer. The water is distributed to the various sections and rises through the tubes to the top of the economizer. The water from the various sections is carried through interconnecting parts at the top of the economizer to the flue-gas inlet end, where it is delivered to the boiler supply line.

Mechanically operated scrapers are moved up and down on the tubes to remove soot accumulations. The motion of the scrapers is auto-

matically reversed when the scrapers reach the tops and the bottoms of the tubes. The tube sections are enclosed by metal or brick sides so that the flue gases are forced to pass through the openings between the tubes and also between the tubes and the enclosing side walls.

Provision is always made for bypassing the flue gases beneath or around the economizer sections when the economizer is out of service.

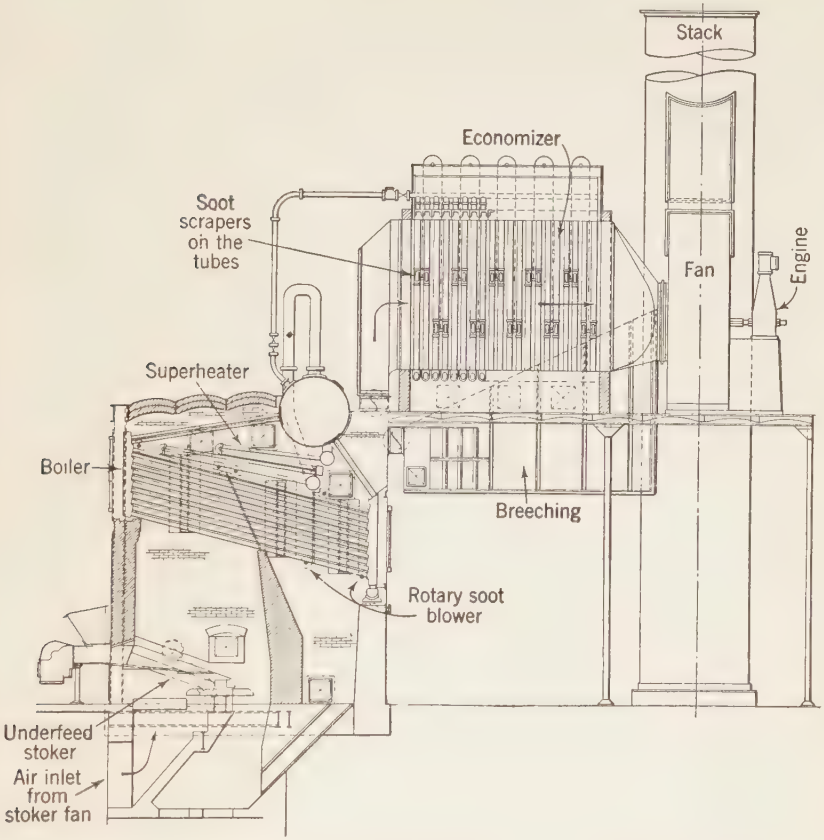


FIG. 107.—Cast-iron tube economizer installed with a Springfield cross-drum water-tube boiler.

Access to the interior of the tubes is secured by removing cover plates from the upper tube headers and hand-hole cover plates from the various interconnecting parts.

140. Steel Economizers.—An integral economizer in conjunction with a Heine V-type water-tube boiler is shown by Fig. 108. The furnace of the unit is arranged to burn either blast-furnace gas, fuel oil, or

powdered coal. The economizer is located in the rear of the boiler setting and consists of an upper and a lower steel drum joined together

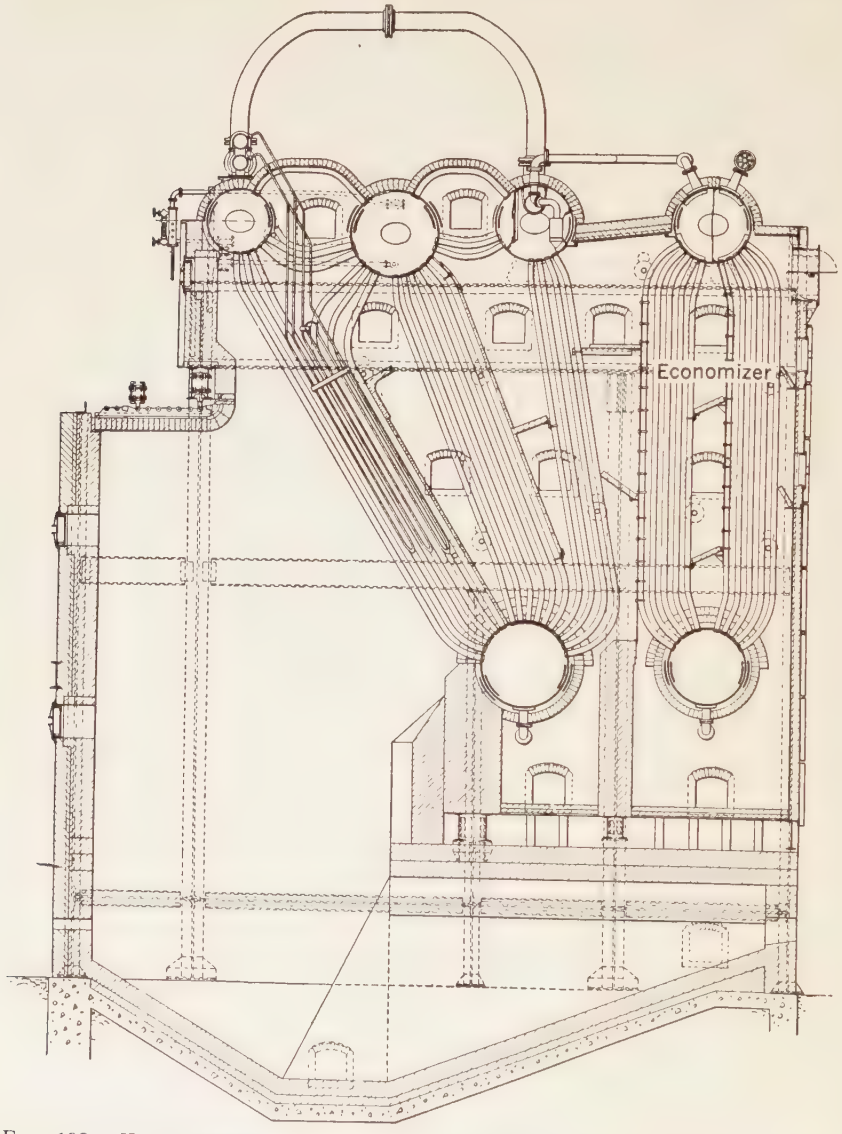


FIG. 108.—Heine 1204-hp. V-type boiler equipped with a 6740 sq. ft. integral economizer.

by two banks of steel tubes. The tubes are expanded into the tube openings of the drums as is the usual custom in steam boilers.

The flue gases after leaving the boiler heating surfaces pass down-

ward around the tubes of the front part of the economizer and thence upward around the tubes of the rear bank to the breeching connection near the top of the rear wall of the setting. Soot and ash accumulations on the economizer tubes are removed by the use of soot blowers, as in the case of the boiler tubes.

The upper drum of the economizer is divided into two compartments by a baffle. The feedwater enters the rear compartment of the upper drum and flows downward through the rear tubes to the lower drum. The flow of water is then upward through the front tube bank of the economizer to the front compartment of the upper drum and then to the rear upper drum of the boiler proper. The water and the flue gases flow in opposite directions, thereby promoting a greater heat transfer owing to the greater temperature differential maintained. The internal parts of the economizer are cleaned by the methods used in the boiler.

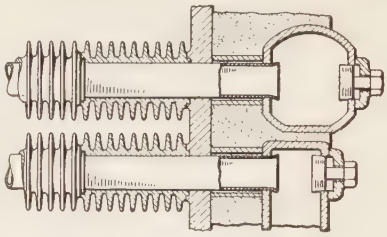


Fig. 109.—Foster steel-tube economizer construction.

Integral economizers similar to that of Fig. 108 are sometimes installed so that the upper drum of the economizer is connected to the rear upper drum of the boiler by tube banks above and below the boiler water line. The lower drum of the economizer under such conditions is not connected to the lower boiler-drum. When they are so installed a steam space as well as a steam disengaging surface exists in the upper drum of the economizer, as it is not entirely filled with water. The feedwater is introduced into the lower drum of the apparatus. Such integral economizers are also known as steaming economizers.

The Foster steel-tube economizer has the ends of its 2-inch tubes expanded into steel headers as illustrated by Fig. 109. The steel tubes are protected against the corrosive action of flue gases by gilled cast-iron rings which fit tightly upon the tubes and which have overlapping joints between the rings. The extended areas of the rings provide additional heat-absorbing surfaces. Access to the interior of the economizer is by means of hand holes in the headers opposite to the ends of the tubes. These openings are closed by means of cover plates as illustrated. A typical method of installing an economizer of this type is illustrated by Fig. 110. The water flow through the economizer is opposite to that of the flue gases. Soot blowers are used to remove ash and soot deposits from the heating surfaces of the economizer. Figure 70 also illustrates an installation of a Babcock and Wilcox steel economizer in connection with an air heater.

141. Regenerative Feedwater Heating.—Modern power stations operate with their auxiliaries of stoker, pumping, draft, and lighting equipment driven by either (1) steam units; (2) steam and electrical motors; or (3) electrical motors.

When all auxiliaries are steam driven a considerable amount of exhaust steam is available in addition to the heat of the flue gases for water heating. Power for electrical drives may be provided by either (1) a house turbine generator unit; (2) the bus bars of the main generating units; or (3) a house generator coupled to the shaft of the main unit.

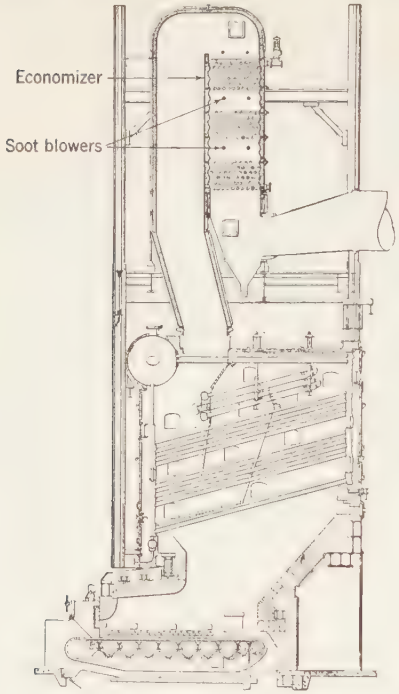


FIG. 110.—Installation of a Foster economizer.

In the latter two cases an independent source of power must be provided for starting the plant. The house turbine generator usually exhausts to a condenser which serves as an open heater. Plants equipped to draw auxiliary power from the main units were so designed with the idea of improving the plant heat balance by regenerative feedwater heating. In any event, by any system of auxiliary drive there may be insufficient exhaust steam for water heating so that steam extraction from the various stages of the turbines may be necessary. It is feasible also to extract steam from the receivers located between the cylinders of compound steam engines.

The Rankine cycle, usually taken as the ideal for the production of steam power, is irreversible, and is not the most efficient way of turning heat energy into mechanical energy. The cycle may be made more nearly reversible and the efficiency improved several per cent by heating feedwater with partially expanded steam extracted from a turbine or an engine. Extracted steam has performed a part of the work which it is capable of doing and when used to heat feedwater displaces an equivalent amount of heat in the boiler. Extraction of steam from various points also relieves the low-pressure stages or cylinders of a considerable volume of expanded steam.

Extraction from more than one stage of a steam turbine is advan-

tageous as it permits a part of the extracted steam to do more work in the prime mover than if the steam were all taken out at one place. Furthermore, extraction at several places provides steam at various pressures and temperatures for the progressive heating of the boiler

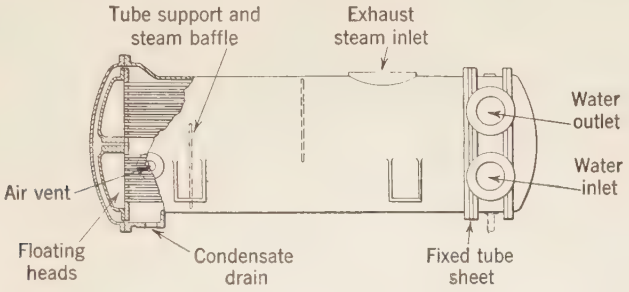


FIG. 111.—Cochrane extraction feedwater heater.

feedwater. The smaller the temperature difference between the steam and the water the more nearly the process approaches reversibility. Usually not more than three or four stages of heaters are profitable.

At pressures near atmospheric, open heaters may be used. Most

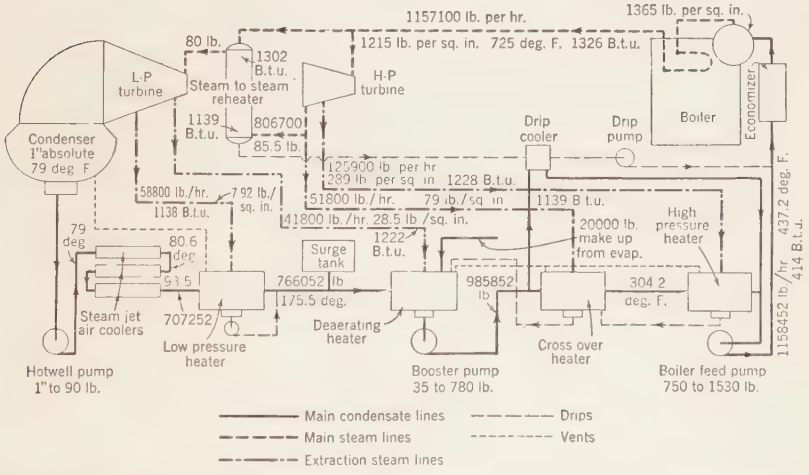


FIG. 112. Flow diagram showing the arrangement and duty of the apparatus of a turbine installation carrying a load of 111,160 kw. at the River Rouge Plant of the Ford Motor Company.

extraction heaters are of the closed type and operate under steam pressures ranging from high to less than atmospheric. Closed heaters of this type must be able to withstand water pressures considerably greater than the steam pressures carried within the boilers. Ample provision

must be made to prevent strains in the heaters by expansion and contraction of the tubes. A Cochrane high-pressure extraction heater with a divided floating tube sheet is illustrated by Fig. 111.

The flow diagram of a high-pressure power plant using a multicylinder steam turbine as a prime mover is illustrated by Fig. 112. This diagram shows four stages of feedwater heaters which utilize steam bled from stages of both the high- and low-pressure turbines. Prior to entering the low-pressure heater the condensate from the turbine condenser is passed through three extraction heaters.[2] Steam at 81.5 lb. per sq. in. is taken from the high-pressure turbine for use in the evaporators which supply 20,000 lb. of make-up water per hour.

PROBLEMS

1. A steam boiler operates at an absolute pressure of 150 lb. per sq. in. and delivers steam of 97 per cent quality. What fuel saving may be expected if a feedwater heater is used to heat the feedwater from 65 to 180 deg. F.?

2. A 500-hp. steam boiler operating at 100 per cent rating delivers steam of 98 per cent quality at an absolute pressure of 415 lb. per sq. in. when the initial feedwater temperature is 85 deg. F. What increase in boiler capacity in pounds of steam per hour may be expected if the feedwater is heated to 200 deg. F. before it enters the boiler? The boiler efficiency and the kind and amount of the fuel used are to be unchanged.

3. Thirty thousand pounds of water per hour were heated in a closed feedwater heater from a temperature of 90 deg. F. to 198 deg. F. The water then passed through an economizer and was heated to a temperature of 350 deg. F., after which it entered a boiler operating at a steam pressure of 225 lb. per sq. in. abs. The boiler delivered steam of 97.3 per cent quality to a superheater, which added 137 deg. F. of superheat. Determine the percentage of the total heat absorbed by the water that was added in (*a*) the feedwater heater, (*b*) the economizer, (*c*) the boiler, and (*d*) the superheater. What was the performance of each piece of equipment in terms of boiler horsepower?

4. An open type of feedwater was used to heat 60,000 lb. of water per hour from a temperature of 70 deg. F. to a temperature of 211 deg. F. Steam of 90 per cent quality was supplied to the heater at an absolute pressure of 16 lb. per sq. in. If the heater losses were 8 per cent of the heat of the steam condensed in the heater how many pounds of steam were used per hour?

5. How many pounds of feedwater will 9000 lb. of exhaust steam, at 20 lb. per sq. in. abs. pressure and 85 per cent quality, heat from a temperature of 65 deg. F. to a temperature of 225 deg. F. in an open type of feedwater heater? Assume the heater losses to be 5 per cent of the heat of the steam.

6. A closed feedwater heater is to heat 8000 gallons of water per hour from a temperature of 50 deg. F. to 280 deg. F. Dry saturated steam at a pressure of 140 lb. per sq. in. abs. is to be used in the heater. Find the number of square feet of heating surface necessary if the coefficient of heat transmission is 350 B.t.u. per sq. ft. per hr. per deg. F.

[2] Heat Engineering, Vol. VII, No. 12, Foster Wheeler Corp.

7. A closed feedwater heater which has 150 sq. ft. of heating surface receives feedwater at a temperature of 56 deg. F. The water is to be heated to a temperature of 185 deg. F. by saturated steam at a pressure of 25 lb. per sq. in. abs. How many pounds of water can be heated per hour if the value of U is 400 B.t.u. per hr. per sq. ft. per deg. F.?

8. A steam boiler plant uses 6000 lb. of coal per hour and discharges 16 lb. of flue gases per pound of coal as fired. The flue gases enter an economizer at 670 deg. F. and leave it at 450 deg. F. Feedwater enters the economizer at 180 deg. F. and leaves at 310 deg. F. How many pounds of feedwater may be heated per hour if the specific heat of the gases is 0.25 and the heat losses from the economizer are 10 per cent of the heat given up by the gases passing through the economizer?

CHAPTER X

DRAFT APPARATUS

CHIMNEYS AND FANS

142. Draft Measurement.—Draft is the difference between the pressure of the air outside and that of the gases within a furnace, a boiler setting, or a chimney. Such pressure differences are variable in amount and are ordinarily of comparatively small magnitude. The measurements are usually made in terms of the height in inches of a water column exerting an equivalent pressure.

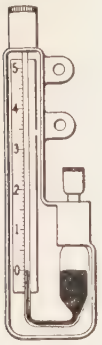


Fig. 113.—Ellison vertical draft gage.

Gages suitable for draft measurements are shown by Figs. 113 and 114. A light oil is used as the gage fluid, and the scale divisions are so placed that the readings obtained are in inches of water equivalent. The use of a gage fluid lighter than water permits the graduation marks to be spaced farther apart so that the readings may be made with a greater degree of accuracy. Inclination of the gage scale further increases the distance between graduation marks and the ease with which the indications may be read.

An Ellison multiple draft gage of the pointer type is illustrated by Fig. 115. The pressures or the pressure differences to be measured are transmitted to the respective cylindrical bell chambers, which are sealed against air or gas leakage by kerosene gage fluid held in the pan at the bottom of the bells. The hollow bell cylinders move up or down respectively for pressures greater or less than atmospheric. The movements thus produced are transmitted by a straight-line motion to the pointer moving over the gage scale. A weight is provided in the linkage mechanism to aid in counter-balancing the gage hand and bell.

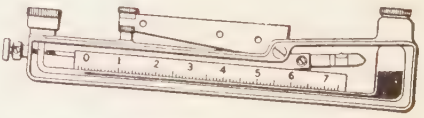


Fig. 114.—Ellison inclined draft gage.

143. Draft Requirements.—Draft is required to produce the flow of the air necessary for combustion through the fuel bed and to promote

the flow of gases from the furnace through the boiler setting and smoke connections into and out of the chimney or stack. The resistances offered to the flow of flue gases vary with (1) the type of boiler, (2) the type of the boiler setting, (3) the arrangement of the baffles within the setting, (4) the length and design of the breeching and smoke connections to the chimney, (5) the accumulations of soot and ashes in the setting and the smoke connections, and (6) the rate at which the boiler is being driven. Information relative to the draft losses through various boilers under operating conditions should be obtained from the builders. The resistances offered to the flow of air supplied to the boiler furnace include air-duct friction and the opposition to air flow produced by the fuel bed. The size of the fuel used, the depth of the fuel bed, and the rate of fuel consumption affect greatly the pressure difference necessary to promote the flow of air into a furnace. Draft requirements[1] for two distinct operating conditions are given by Table XVI and Fig. 116.

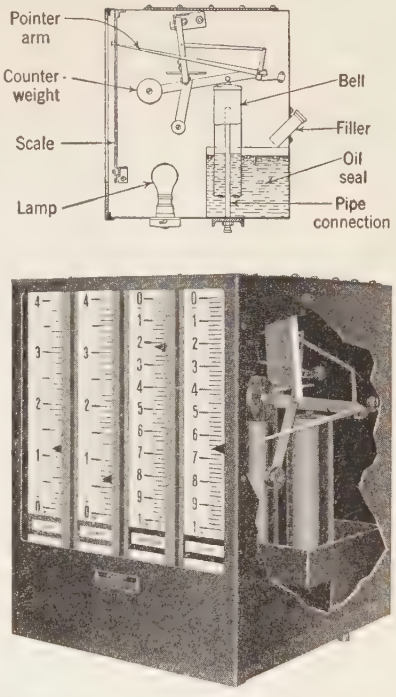


FIG. 115.—Ellison pointer draft gage.

Anything which offers resistance to the flow of air into and through the fuel bed, and the flow of flue gas from the furnace to the chimney, increases the amount of draft that must be available. No fixed amount can be applied to all installations, as each boiler plant must be considered individually.

144. Draft Production.—Draft is produced by either of two distinct methods, and is classified as *natural* and *mechanical*.

145. Natural Draft.—The pressure difference thus secured is produced solely by the use of a stack or chimney. The magnitude of the force of draft is dependent upon the height of the chimney top above the level of the furnace grates, and the average temperature difference between the flue gases within the chimney and the outside air. Weather variations, boiler-operating conditions, and the chimney height have a

[1] "Furnace Operation Shown by Draft Measurement," *Power Plant Engineering*, January 1, 1928, p. 15.

marked effect upon the amount of draft that may be secured by the use of a particular chimney.

TABLE XVI

Typical Conditions in Natural Draft Furnace
Draft Required in Furnace for Various Rates of Combustion

Combustion Rate per Square Foot of Grate in Pounds per Hour	Draft in Inches of Water	
	For Bituminous	For Anthracite
5	0.02 to 0.04	0.06 to 0.16
10	0.05 to 0.10	0.16 to 0.40
15	0.07 to 0.15	0.30 to 0.75
20	0.10 to 0.25	0.45 to 1.25
25	0.12 to 0.32	0.65
30	0.17 to 0.45	0.90
35	0.21 to 0.58	
40	0.27 to 0.75	
45	0.32 to 0.95	

Natural draft does not require the expenditure of mechanical power in its production and is suitable for small plants when the load condi-

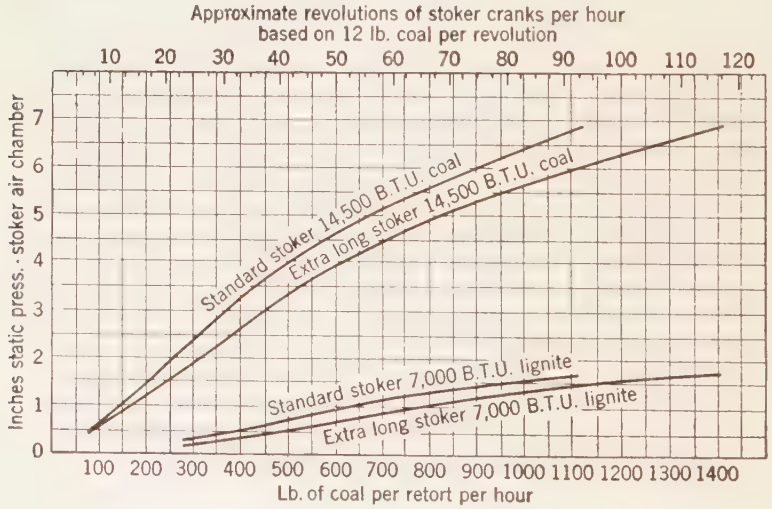

Approximate revolutions of stoker cranks per hour
based on 12 lb. coal per revolution
10 20 30 40 50 60 70 80 90 100 110 120
Inches static press. - stoker air chamber
0 1 2 3 4 5 6 7
Standard stoker 14,500 B.T.U. coal
Extra long stoker 14,500 B.T.U. coal
Standard stoker 7,000 B.T.U. lignite
Extra long stoker 7,000 B.T.U. lignite
100 200 300 400 500 600 700 800 900 1000 1100 1200 1300 1400
Lb. of coal per retort per hour

Fig. 116.—Approximate forced-draft pressures for underfeed stokers.

tions are not extremely variable, or where the boilers are not required to develop great capacities quickly.

146. Chimneys and Stacks.—Either of these structures is used with all draft systems for conveying the smoke or flue gases to an elevation above that of surrounding objects and buildings. Structures for the disposal of flue gases or the production of natural draft, when constructed of brick, concrete, or radial brick, are known as chimneys; those built of sheet steel or steel plates are designated as stacks. Stacks built of the lighter sheet metal require the use of supporting guys; those built of heavier steel plates may be made self-supporting. An illustration of a typical radial-brick chimney is shown by Fig. 117.

Brick and tile chimneys are likely to have leakages due to poor bonding of the materials or cracks due to settlement of the structure. Concrete chimneys are also subject to cracking. Air leakages into a chimney producing natural draft reduce the amount of the draft available at the smoke connection. Steel stacks, unless kept carefully painted, are subject to the corrosive action of the air and weather. Unlined steel stacks are corroded also by the constituents of the flue gases, so that the life of an unlined stack is much shorter than that of a chimney built of bricks, radial bricks, or concrete.

147. Chimney Height and Area.—Numerous empirical rules have been developed for the calculation of the internal dimensions of chimneys and stacks used for the production of natural draft. The empirical sizes thus obtained may prove adequate in some cases and inadequate in others. A theoretical method of chimney design, in the case of natural draft, is based on an estimation of the probable total draft requirements, the probable temperatures within and without the chimney, and the volume of the flue gases to be handled per hour.

Natural draft is produced by the difference in the weight of a column of hot gas within the chimney and that of a cold air column of equal height and cross-sectional area without the chimney. Therefore, the higher the chimney, or the greater the temperature difference, or both, the greater is the force produced.

The height of a chimney necessary to produce a given theoretical draft at its base may be calculated in the following manner. A chimney AB, Fig. 118, has a height of h feet and is filled with hot flue gases whose average temperature is t_c deg. F. Outside the chimney is a column of cool air whose height and cross-sectional area are equal to those of the chimney and whose average temperature is t_a deg. F. The outside air column is heavier than the hot gas column, consequently the two are not in equilibrium. Hence, there is a movement of the cool air to replace the lighter heated gases. If the two columns were brought into equilibrium, it would be necessary to increase the height of the hot column by an additional amount equal to h_c. The existing unbalanced condi-

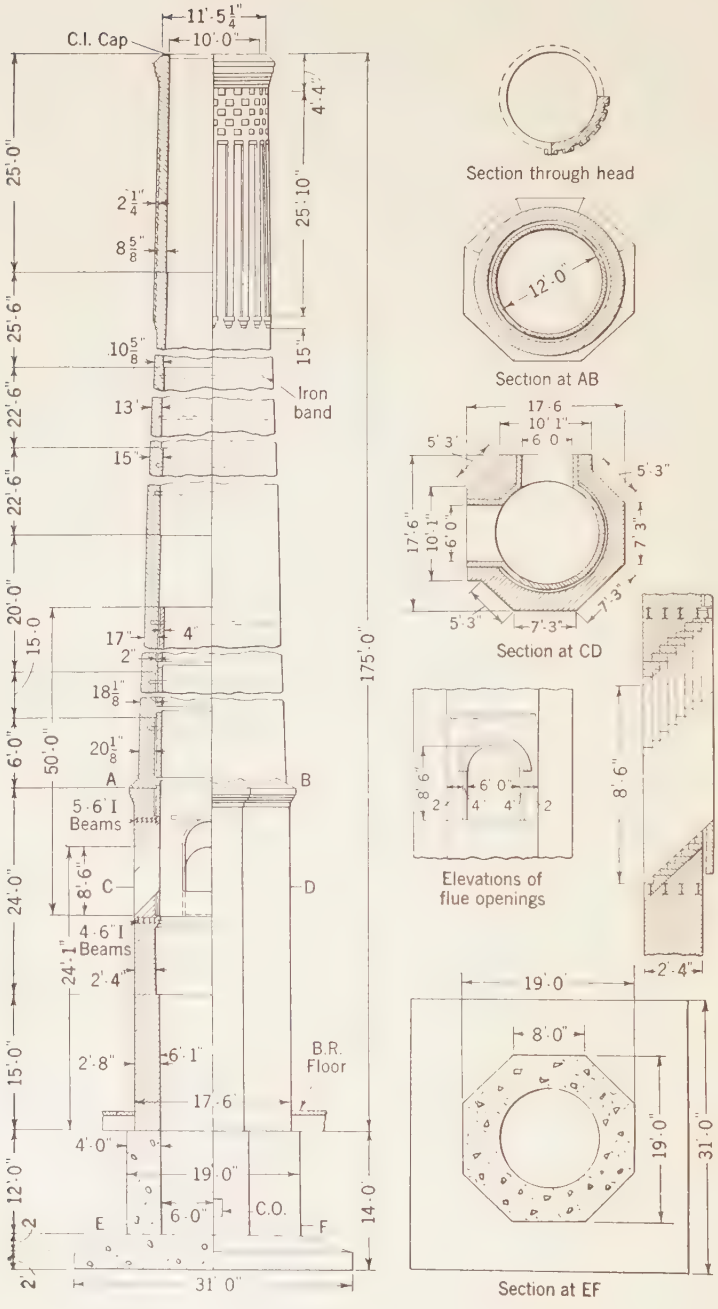


FIG. 117.—Radial-brick chimney, University of Illinois power plant.

tion promotes the flow of gases within the chimney, and h_c is a measure of the force of the draft.

If equilibrium were established the following relations would exist for volumes of equal cross-sectional areas, and the volumes Q would be proportional to the absolute temperatures of the gases:

$$Q : (Q + Q_c) = h : (h + h_c) = (460 + t_a) : (460 + t_c)$$

$$h(460 + t_c) = (h + h_c)(460 + t_a)$$

$$h_c = \frac{h(t_c - t_a)}{460 + t_a}. \tag{94}$$

The unbalanced head h_c is expressed in terms of feet of head of the hot gases so that it is desirable to establish a relation between h_c and the theoretical draft h_w measured in inches of water. When the density of the flue gases d and the density D of the draft gage fluid are expressed in pounds per cubic foot, h_c may be written

$$h_c = \frac{h\,(t_c - t_a)}{460 + t_a} = \frac{h_w D}{12d},$$

$$h = \frac{h_w\, D\,(460 + t_a)}{12d\,(t_c - t_a)}; \tag{95}$$

$$h_w = \frac{h\,(t_c - t_a)\,12d}{D\,(460 + t_a)}. \tag{96}$$

Actually, the chimney is not likely to produce the theoretical draft, because of leakage and losses from friction and velocity of flow. A rough approximation of the actual draft may be 0.80 of h_w.

The cross-sectional area of the chimney is calculated by using the volume of flue gases produced per second and the actual velocity of the gases. The theoretical velocity V, in feet per second, produced by the unbalanced head h_c, may be expressed by $V = \sqrt{2gh_c}$, in which g equals the acceleration due to gravity, 32.17 ft. per sec. per sec.

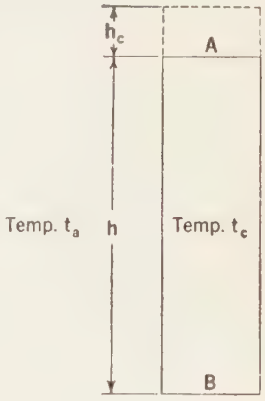


FIG. 118. — Elementary chimney.

$$V = \sqrt{2 \times 32.17}\,\sqrt{\frac{h(t_c - t_a)}{460 + t_a}} = 8.02\,\sqrt{\frac{h(t_c - t_a)}{460 + t_a}}. \tag{97}$$

Under actual conditions of chimney operation the flue-gas velocities are only 30 to 50 per cent of the theoretical velocities, owing to roughness of the interior surfaces of the chimney, etc. The cross-sectional area of the chimney is

$$A = \frac{Q}{V \times K'} \tag{98}$$

in which A = chimney area, sq. ft.
Q = volume of gases handled, cu. ft. per sec.
K = coefficient of velocity, 0.3 to 0.5.
V = theoretical gas velocity, ft. per sec.

Example.—A steam boiler plant consumes 20,000 lb. of coal per hour. The weight of dry flue gases formed per pound of coal fired is 20 lb. The gases enter the chimney at a temperature of 650 deg. F., and the average chimney gas temperature is 500 deg. F. A chimney is desired that will produce a theoretical draft of 0.9 in. of water at its base when the outside air temperature is 90 deg. F. What must be the dimensions of the chimney?

Solution.—

$$h = \frac{0.9 \times 62.11\ (460 + 90)}{12 \times 0.04138\ (500 - 90)} = 151 \text{ ft.}$$

$$Q = \frac{20{,}000 \times 20}{0.0359 \times 60 \times 60} = 3095 \text{ cu. ft. per sec.}$$

$$A = \frac{3095}{8.02 \times 0.4\sqrt{\dfrac{151\ (500 - 90)}{460 + 90}}} = 91 \text{ sq. ft.}$$

$$\sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times 91}{3.1416}} = 10.73 \text{ ft. dia.}$$

The densities of flue gases may be taken, with small error, as those of dry air at the same temperature and pressure, and were so used in the foregoing example. Gas or air densities are readily calculated by the use of equation 15, $PV = MRT$. See also Table XXIV.

148. Kent's Chimney Formula. One of the empirical equations frequently used in chimney design is that proposed by William Kent. This formula is based on a coal consumption of 5 lb. per boiler hp. per hr., and is

$$\text{Boiler hp.} = 3.33 A_e \sqrt{h}. \tag{99}$$

A_e = effective area of the chimney, sq. ft. An allowance for a dead gas space 2 in. in thickness around the inside of the chimney walls is made.

$A_e = A - 0.60\sqrt{A}$, where A = the actual chimney area, sq. ft.
h = chimney height, ft.

149. Mechanical Draft.—Draft produced by the action of steam or air jets or centrifugal fans and blowers is termed *mechanical draft.* These methods of production are used where positive and certain draft must be maintained irrespective of weather or boiler-operating conditions and where the natural draft provided by stacks or chimneys is deficient. Auxiliary boiler equipment such as economizers and air preheaters offers resistance to the flow of flue gases as well as reduces the flue-gas temperatures. In such cases a chimney necessary to produce the required amount of draft may be excessive in height and cost, so that mechanical draft is necessary. In general, mechanical draft is desirable when boilers are to be operated at capacities greatly in excess of their rating or when they must quickly supply sudden demands for steam.

Boilers provided with mechanically produced draft do not require stacks or chimneys so high or so expensive as those necessary for natural draft. In some cases where mechanical draft is used, chimneys of considerable height are built to prevent the smoke and flue gases, together with the cinders and fine ashes that may be discharged, from being objectionable to adjacent property owners. The initial cost, maintenance, depreciation, and operating costs of mechanical draft equipment must be balanced against similar costs of stacks or chimneys necessary to produce the same amount of draft when a selection is to be made between the two systems.

150. Mechanical Draft Systems.—Two systems are in common use. These are *forced* draft and *induced* draft. Either system may be used alone or in combination as illustrated by Fig. 119.

Forced draft is secured by blowing air into a closed boiler room, into chambers or air boxes under stokers or grates, or through pulverized-fuel burners. Air is supplied under pressure and thus forces its way through the fuel bed or burner into the furnace. The general practice in forced draft is to remove the products of combustion from the boiler setting by natural or induced draft or a combination of the two. The air supply fans or blowers do not force the products of combustion out of the stack.

Induced draft is procured by a jet blower or a centrifugal fan located in the smoke connection between the boilers and the stack or at the base of the stack. When an economizer is installed the induced-draft equipment is placed between the outlet end of the economizer and the stack. The action of induced-draft apparatus is to reduce the pressure of the gases within the boiler setting below that of the atmosphere. The gases are discharged into the stack under a positive pressure.

Jets of steam discharged through the nozzles of aspirating devices

located within the stack near the smoke connection will produce induced draft. Ordinarily the use of such equipment with live steam is not to be recommended, as it is not economical in operation.

Induced-draft appliances may produce a pressure within the furnace well below that of the atmosphere so that the outside air pressure is sufficient to force the requisite amount of air through the fuel bed. In the

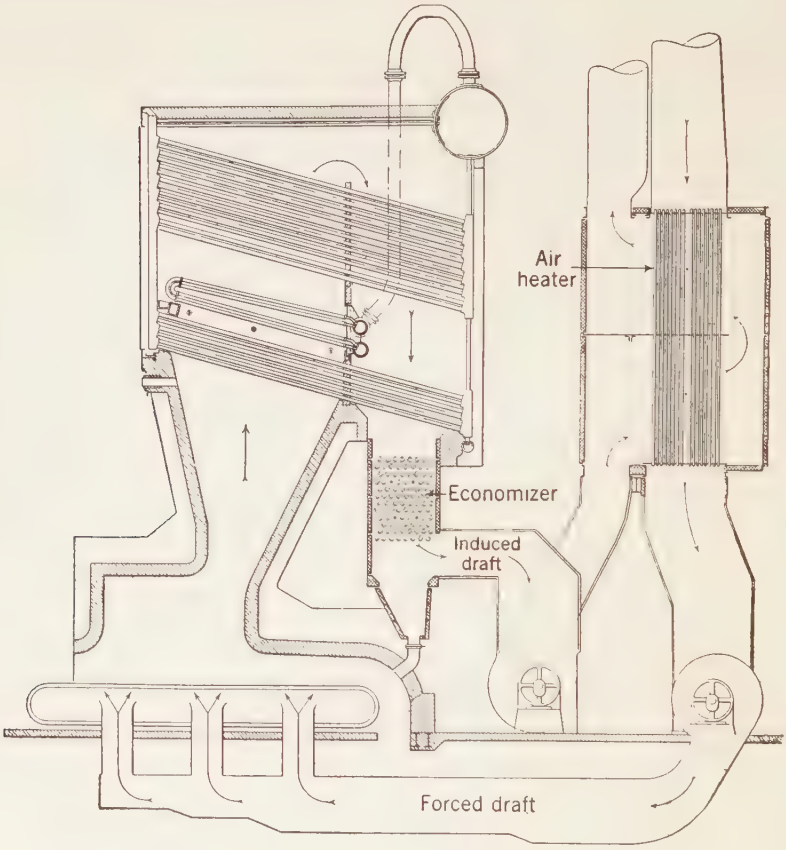


Fig. 119.—Forced- and induced-draft installation.

case of underfeed stokers, forced draft is required for the air supply. When a combination of forced and induced draft is used with practically atmospheric pressure in the furnace above the fire, the term *balanced draft* is applied.

151. Evasé Stacks.—The streamline stack for the production of induced draft has been used to a considerable extent in Europe and is finding usage in this country. Three types of these stacks are

illustrated by Fig. 120. The type 1 stack is suitable for drafts greater than 1.5 in. of water and the type 3 stack for drafts less than 1.5 in. of water. Either type of stack may be operated to give full induced draft or natural draft without the fan in operation. Where corrosive or very hot gases are handled, type 2 stacks, which have the fan located externally, may be used; in this case the fan blows a jet of cool air through the ejector nozzle. Type 1 stacks are built with single and twin fans.

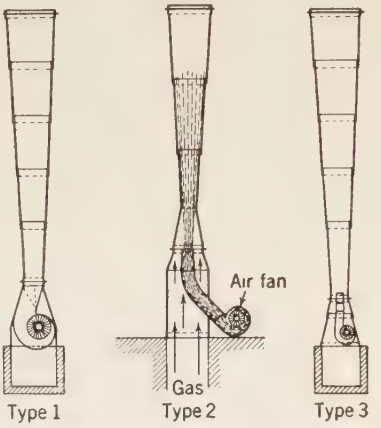


FIG. 120.—Prat-Daniel streamline stacks.

The stack sides form an angle of about 7 deg. with the vertical, and an ejector action is secured as the gases pass at high velocities through the constricted section. The reduction of the velocity of the gases is gradual in the flared part of the structure, and a considerable amount of resistance to gas flow is eliminated. A saving in the amount of power required to handle the gases is effected. The stacks are comparatively light in weight and usually have very short smoke-pipe connections to the boilers. A typical installation of Evasé stacks with economizers, air preheaters, and forced-draft fans is shown by Fig. 121.

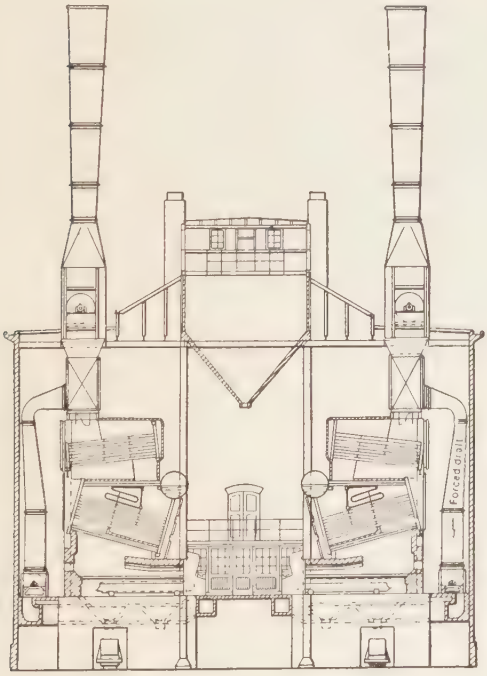


FIG. 121.—An installation of Evasé or streamline stacks.

152. Centrifugal Fans.—The air- or gas-moving equipment of either forced- or induced-draft systems may employ centrifugal fans of the following types: (1) steel-plate, and (2) multiblade. Either type or both types may be used.

All fans consist of a wheel which is rotated within a housing usually constructed of sheet metal. The housing of steel-plate and multivane fans is of a scroll shape, as in Fig. 122, that permits the air or gas to be carried away from the periphery of the wheel with slight disturbances and power losses.

Fan wheels have some sort of vanes or blades located at or near the wheel rim. The action produced by a wheel in rotation is secured as a result of the tendency of air, adjacent to the forward sides of the fan blades, to flow radially outward. This action is due to centrifugal

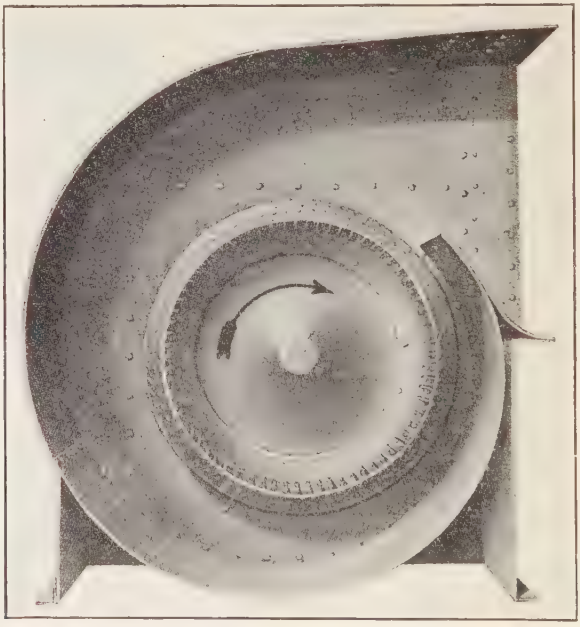


Fig. 122.—New York Blower Company fan wheel and scroll.

force, and the air is discharged from the tips of the blades into the scroll of the housing. As a result of this movement a pressure less than atmosphere is produced at the center of the wheel and a positive air pressure is built up in the scroll. Air or gas flows axially into the wheel to take the place of that discharged. Consequently, not only may centrifugal fans be used to exhaust air or gases through a duct system attached to the fan-housing inlet, but also they may be made to discharge the same air or gases under several inches of water pressure through a duct system leading away from the housing scroll.

Steel-plate Fans.- In fans of this type the wheels consist of one or two spiders, each having from 6 to 12 arms. Each arm or pair of arms

carries a flat steel float or blade of some radial length as in the typical wheel of Fig. 123. The wheel floats may be straight, or curved either forward or backward, depending upon the operating characteristics desired. The blade curvature has a marked effect upon the performance of a given type of fans.

Multiblade Fans.—Such fan wheels are built up of two or more annular rings with many narrow curved blades inserted between them as illustrated by Figs. 122 and 124. The blades may be curved forward or backward, depending upon the performance characteristics desired.

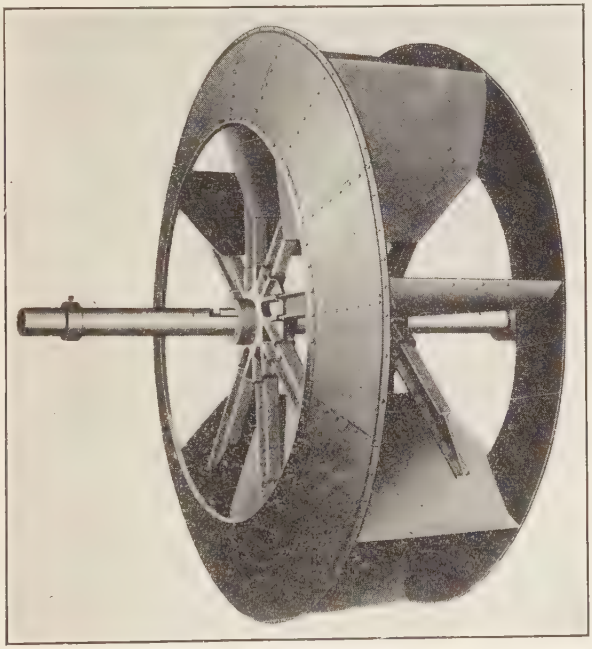


Fig. 123.—American Blower Corporation steel-plate fan wheel.

Multiblade wheels designed for high speeds and pressures are reinforced by a number of annular rings, and the blades are made short laterally, in order to secure greater rigidity. Figure 124 illustrates modern high-speed fan wheels with backward-curved blades. As shown, two single inlet wheels are mounted upon a shaft used within a double inlet housing. Wheels of this type give high volumetric efficiencies and have a wide range of capacity at constant speed with small changes in the power requirements.

Fans having wheels as shown by Fig. 125 are of the multiblade type, and the term *conoidal* is applied to the blades which are in the form of

the surfaces of two tangent cones. This design gives the blade a forward curve at its inward edge and a marked backward curvature at the periphery or exit edge. A blade producing desirable operating character-

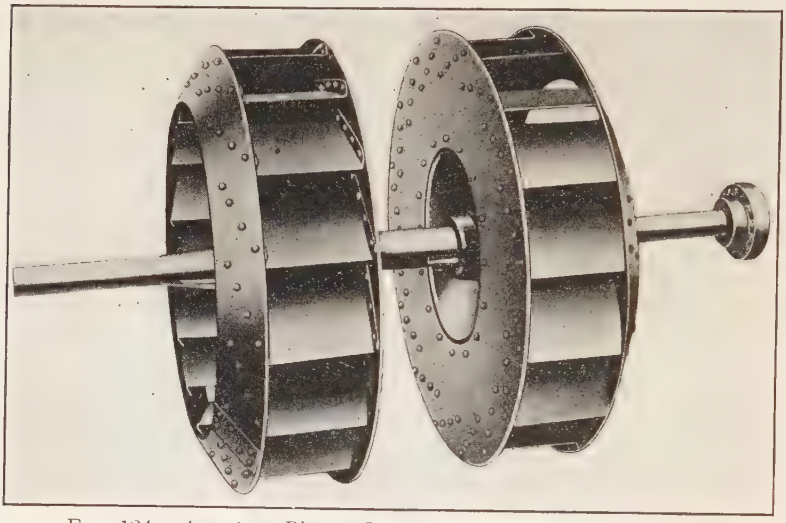


Fig. 124.—American Blower Corporation high-speed fan wheels.

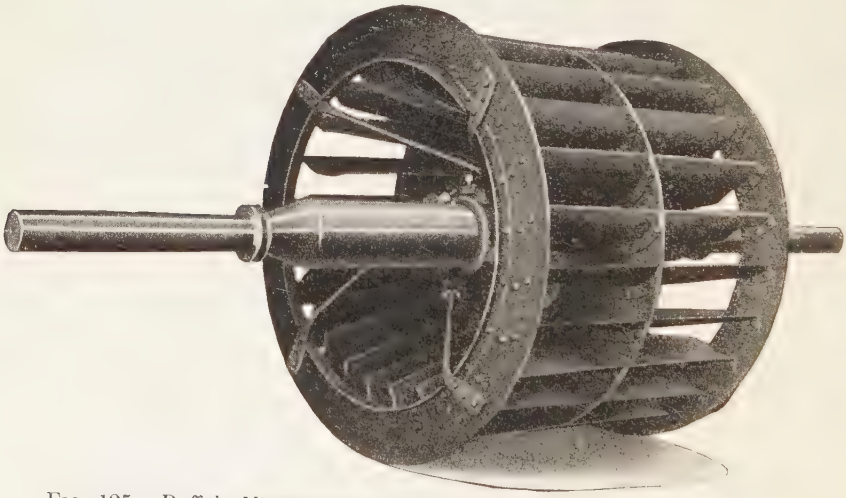


Fig. 125.—Buffalo Niagara conoidal, double-inlet, double-width fan wheel.

istics is secured which also has sufficient rigidity without the use of stiffening rings.

The name *radial-flow* is given to the wheel of Fig. 126. The backward curvature of the blades is such that the air leaves them quietly

with a minimum of shock losses. These fans operate with a characteristic that shows a decrease in the power necessary to drive them with an increase in capacity after a maximum power requirement has been reached.

High-pressure blowers are used only for the production of forced draft. Steel-plate fans, from the nature of the design, are best suited for slow rotative speeds and may be directly connected to steam engines so that speed can be controlled by manipulation of the engine throttle. Steel-plate fans may be driven through belt connections from high-speed electric motors when the proper selection of a ratio between driving and driven pulley diameters is made. In such cases, the speed may be varied by use of speed-control devices for the motor, but these add materially to the initial cost of the system.

Fans of the conoidal, radial flow, and backward-curved multiblade types are high-speed machines suitable for direct connection to either steam turbines or electric motors. In general, the high-speed fans are more efficient than the older and slower-speed steel-plate fans.

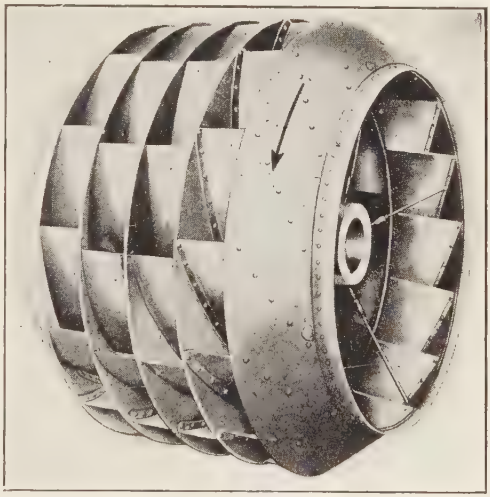


FIG. 126.—Green double-inlet radial-flow fan wheel.

Fans of any of the foregoing classes, when used to produce induced draft or forced draft with preheated air, should be equipped with water-cooled bearings on account of the temperatures of the gases or air handled. Induced-draft fans of all types should be of rugged construction because they are subjected to the corrosive action of the flue gases and the abrasion of ash and cinder particles discharged.

The power requirements of induced-draft fans handling a given weight of flue gases are greater than those of fans handling the same weight of cool air against the same total pressure in a similar time period.

153. Propeller Fans.—Units of the type shown by Fig. 127 are used in forced-draft work either in connection with stokers or with hand-fired boiler furnaces where they may be used to force air under pressure into closed ashpits. Figure 128 illustrates the use of a propeller fan in con-

nection with a hand-fired boiler furnace. Such units are often referred to as under-grate blowers. Fan wheels of the propeller type are mounted in cylindrical casings and may be driven by either steam turbines or electric motors. The Wing blower shown in Fig. 127 is driven by a steam turbine. Such blower units may discharge through very short outlet ducts and are not provided with inlet ducts.

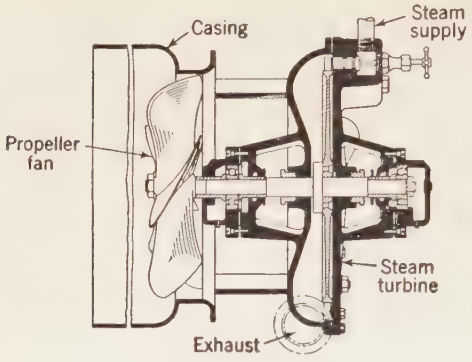


Fig. 127.—Wing blower.

154. Fan Nomenclature.—Fans of the steel-plate and multiblade types are designated according to the (1) number of inlets—single or double; (2) width of the wheel, single or double; (3) discharge—top, bottom, vertical, horizontal, or angular; (4) housing—full, seven-eighths, or three-quarters; and (5) right or left hand.

A full-housed fan is one in which the fan scroll is completely above the base upon which the fan rests, as illustrated by Fig. 122. Seven-eighths or three-quarters full-housed fans have the scroll extending below the top of the supporting base. More of the scroll is located below the base line in the three-quarters full-housed than in the case of the seven-eighths full-housed fan. Each builder has his own method of designating fans as right or left hand so that the manufacturer's catalog should be consulted as to the hand of any particular fan.

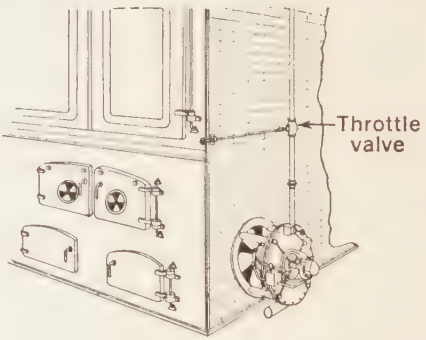


Fig. 128.—Hand-fired boiler with under-grate blower.

155. Fan Pressures.—Fans in operation maintain a *total pressure* which is made up of two components, *velocity pressure* and *static pressure*. Under certain conditions of operation either of these two components may be equal to zero, but both are never equal to zero when the fan is running.

Velocity pressure is utilized in creating and maintaining the velocity of air or gas flow. *Static pressure* is utilized in overcoming frictional and other resistances offered to the flow of air or gases. Static pressure,

velocity pressure, and total pressure are all interrelated. Thus, if the gas handled has a velocity increase in the duct system, a part of the static pressure available at that particular locality is used to create the additional velocity pressure necessary. Likewise, if the speed of flow is reduced at some point, a part of the velocity pressure at that point will be converted back into static pressure.

Any fan which develops a large static pressure in comparison with its total pressure has a greater capacity to overcome the resistances offered to air or gas flow, by duct friction, boiler-setting baffles, or fuel-bed obstructions. For steel-plate fans the most economical operation is secured when the ratio of the total to the static pressure is between

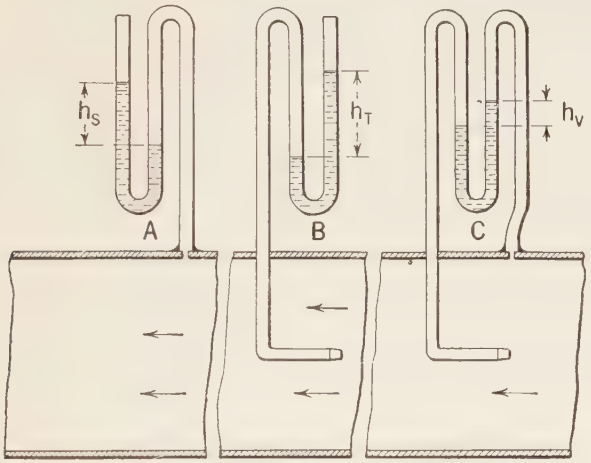


Fig. 129.—Air pressure measurements—greater than atmospheric pressure.

1.2 and 1.3; for multiblade fans the most satisfactory ratio is between 1.05 and 1.15.

A fan creates pressures greater than atmospheric in a duct on the discharge side, and generally produces pressures less than atmospheric in a duct on the inlet side; therefore consideration must be given to pressure measurements in both outlet and inlet ducts.

156. Pressures Greater than Atmospheric.—Figure 129 illustrates a duct of circular section in which air is flowing at some velocity with a static pressure above that of the air outside the duct. The static pressure existing within the duct at any location may be obtained at that point by attaching a draft gage or manometer of sufficient capacity to a connection made at right angles to the longitudinal axis of the duct as at *A*, Fig. 129. The gage connection should not extend through the duct wall, and the small opening should be free from burrs or irregu-

larities. Owing to the fact that turbulence of flow may exist in the duct, it is likely that the static pressure will vary at different points about the circumference of the duct section. Therefore, it is desirable that several static-pressure openings in the duct at the plane of the section be joined by a piezometer ring, Fig. 130, to the manometer. With such an arrangement the average static pressure may be more accurately secured.

The velocity pressure is measured by the determination of the total pressure and the subtraction of the static pressure from it, either mathematically or by the proper attachment of the static and total pressure connections to the indicating gage. If a piece of tubing of small internal diameter and thin walls is bent at right angles and has the end of the short leg cut squarely across and the tube walls ground to a thin edge, a device known as a Pitot tube is formed. The tube is inserted into the duct at *B*, Fig. 129, with the bent portion of the tube located parallel to the longitudinal axis of the duct, so that the moving air will flow directly against the open end of the tube. When the outer end of the tube is attached to a manometer the impact of the air, as well as the static pressure, at that particular location will be indicated. The reading so obtained is the **total pressure**. When the velocity pressure alone is desired the total and static pressure attachments are made to a single gage as at *C*, Fig. 129.

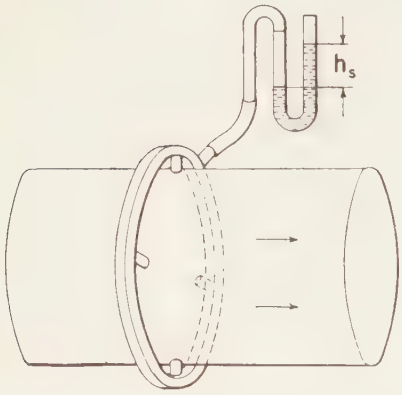


FIG. 130.—Piezometer ring.

A commercial type of Pitot tube as adopted by the American Society of Heating and Ventilating Engineers, Fig. 131, combines in a single instrument both static and total pressure openings. Pitot tubes are not adapted to the measurement of velocity pressures corresponding to gas flows of less than 6 lin. ft. per sec. unless used with very sensitive and delicate pressure-measuring gages.

Velocity-pressure measurements should be made in a straight section of pipe at least 20 duct diameters in length, with at least 10 duct diameters of straight pipe on either side of the Pitot tube station. Erroneous or variable data may be secured when the observations are taken near a fan outlet or a bend in the duct.

157. Pressures Less than Atmospheric. When the pressure within a duct is less than atmospheric, as in the suction inlet of an exhaust

fan, the measurements of the static pressure h_s and the total pressure h_t are as indicated by A and B of Fig. 132. Under such conditions the

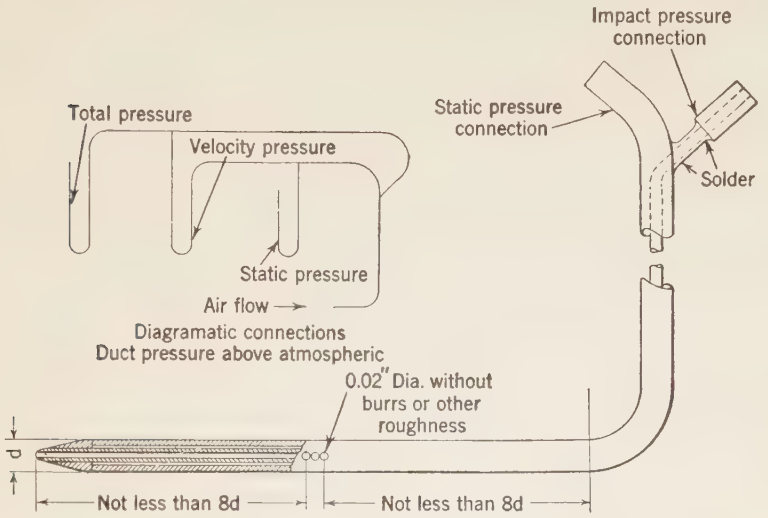


Fig. 131.—Pitot tube.

gage readings should be given a negative sign to indicate that the measurements taken are so many units of pressure less than the atmos-

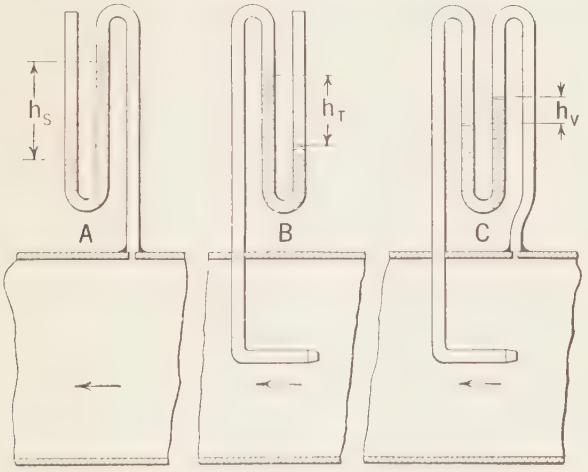


Fig. 132.—Air pressure measurements—less than atmospheric pressure.

pheric pressure. Because the fan wheel produces a partial vacuum at its wheel center, the absolute pressures within the duct are less than the pressure of the atmosphere outside of the inlet duct.

In Fig. 132 let the static pressure measurement h_s be -1.25 in. of water and the total pressure measurement h_t be -1.00 in. of water, when the barometric pressure is 29.92 in. of mercury or $29.92 \times 13.6 = 406.91$ in. of water. The absolute static pressure is $406.91 + (-1.25) = 405.66$ in. of water, and the absolute total pressure is $406.91 + (-1.00) = 405.91$ in. of water. The velocity pressure h_v as indicated by the gage at C in Fig. 132 is the total pressure minus the static pressure. This difference may be taken between the absolute pressures as $(405.91 - 405.66) = 0.25$ in. of water or as the difference of the gage pressures $-1.00-(-1.25) = 0.25$ in. of water.

158. Total Pressure Difference Developed by a Fan.—Fans operate with the gases handled arriving at the inlet to the fan housing under atmospheric pressure when no inlet duct is used, and at pressures less than atmospheric when exhausting through an inlet duct. Certain types of blowers such as gas boosters have pressures greater than atmospheric in the inlet duct to the blower or fan.

When the fan has no suction duct the entry losses to the fan housing are considered as a part of the fan losses and are reflected in the mechanical efficiency of the fan. If the fan has no discharge duct leading away from its outlet the discharge static pressure is zero and the total pressure at the discharge outlet is equal to the average velocity pressure at the fan outlet. In any event, the average total pressure difference (total pressure) created by a fan is the average absolute total pressure at the fan outlet *minus* the average absolute total pressure at the fan inlet.

Example.—A fan having no inlet duct discharges air into a duct at 1.00 in. of water static pressure and an average velocity pressure of 0.15 in. of water when the barometric pressure is 29.92 in. of mercury or 406.91 in. of water. Find the total pressure difference developed by the fan.

Solution.—

$$h_T = [406.91 + (1.00 + 0.15)] - [406.91 + 0]$$
$$= 1.15 - 0 = 1.15 \text{ in. of water.}$$

Example.—A fan maintained at its outlet an average static pressure of 1.25 in. of water with an average outlet velocity pressure of 0.35 in. of water. In the suction duct near the fan inlet the static pressure was -1.25 in. of water and the velocity pressure was 0.25 in. of water. Barometric pressure 29.92 in. of mercury or 406.91 in. of water. Find the total pressure difference developed by the fan.

Solution.—

$$h_T = [406.91 + (1.25 + 0.35)] - [406.91 + (-1.25 + 0.25)]$$
$$= 1.60 + 1.00 = 2.60 \text{ in. of water.}$$

Example. A blower received gas at $+0.25$ in. of water static pressure and a velocity pressure of 0.35 in. of water in the inlet duct. The discharge static

pressure was 15 in. of water with a velocity pressure of 0.75 in. of water. Find the total pressure difference produced by the blower. Barometer 29.92 in. of mercury.

Solution.—

$$h_T = [406.91 + (15.0 + 0.75)] - [406.91 + (0.25 + 0.35)]$$
$$= 15.75 - 0.60 = 15.15 \text{ in. of water.}$$

The absolute pressures were used in the solutions of the examples solely for the purpose of taking the difference between two pressures both of which were expressed positively above the same base of reckoning. Inspection of the solutions indicates that the barometric pressure is canceled in all cases. Therefore, the total pressure difference h_T may be calculated from the gage readings alone if proper attention is given to the algebraic signs.

$$h_T = (\textit{average total pressure at fan discharge, inches of water}) - (\textit{average total pressure at fan inlet, inches of water}). \quad (100)$$

159. Velocity of Flow.—The velocities of gas flow may be readily computed from measured velocity pressures by the equation

$$V = \sqrt{2g\frac{Dh_v}{12d}}, \quad (101)$$

in which V = velocity, ft. per sec.
g = acceleration due to gravity, 32.17 ft. per sec. per sec.
D = weight of 1 cu. ft. of water at the temperature of the gage fluid. Table XXI.
h_v = velocity pressure measured, in. of water.
d = density of the fluid flowing, lb. per cu. ft.

Example.—Find the velocity of flow corresponding to a velocity pressure of 0.25 in. of water when air at 70 deg. F. is being handled. Barometer, 29.92 in. of mercury. Temperature of the gage fluid, 70 deg. F.

Solution.—

$$V = \sqrt{\frac{2 \times 32.17 \times 62.3 \times 0.25}{12 \times 0.075}} = 33.4 \text{ ft. per sec.}$$

The velocity pressure measured by the use of a Pitot tube is the one existing at the open end of the tube. The velocity of flow is not uniform over a duct cross-section so that several velocity-pressure determinations are required to secure the average velocity of flow existing at any Pitot tube station. The duct cross-section is usually divided into a number of equal areas, and observations are made with the tube properly located within these areas. A scheme of locating a tube at various points on two diameters of a circular duct cross-section and in a rectangular or

square duct is illustrated by Fig. 133. Velocity-pressure observations made at points on diameters that are at right angles to each other are used for the calculation of the average velocity of flow in the circular duct. The square root of the velocity pressure is used in equation 101; therefore, the average square roots of the velocity pressures taken should be computed and not the average of the velocity pressures read. Thus

$$\sqrt{h_{v_a}} = \frac{\sqrt{h_{v_1}} + \sqrt{h_{v_2}} + \sqrt{h_{v_3}} \ldots . + \sqrt{h_{v_n}}}{n}, \qquad (102)$$

where n is equal to the number of observations taken.

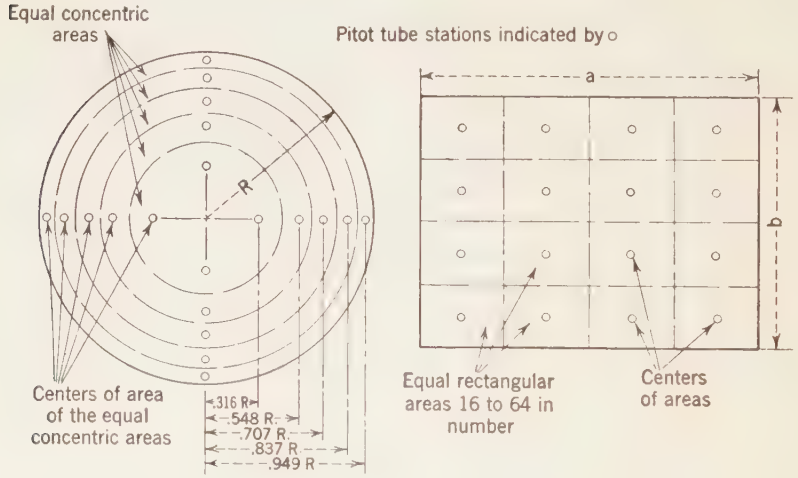


Fig. 133.—Locations of a Pitot tube for a duct traverse.

Knowledge of the relation of the average velocity in the duct to the velocity existing at the duct center is desirable. The center velocity is usually but not always greater than the average velocity of flow. The ratio is known as the *center coefficient* and is expressed as

$$C = \frac{V_a}{V_c} = \frac{\sqrt{h_{v_a}}}{\sqrt{h_{v_c}}}, \qquad (103)$$

where C = center coefficient.
V_a = average velocity in duct, ft. per sec.
V_c = velocity at the center of the duct, ft. per sec.

The center coefficient varies with the velocity of flow, the density of the gas handled, and the shape of the duct cross-section. Usual values of C for ducts of circular shape range from 0.88 to 0.93, but ducts of other

shapes may have coefficients greater or less than these values. When the center coefficient C has been accurately determined, the center velocity pressure h_{v_c} only need be measured, and $V_a = CV_c$.

160. Volume of Discharge.—The volume of gas flowing is equal to the product of the duct section in square feet and the average velocity of flow at that section.

$$Q = AV_a. \tag{104}$$

Q = cu. ft. discharged per sec.
A = duct cross-section area, sq. ft.
V_a = average velocity of flow, ft. per sec.

161. Fan Air Horsepower.—The power output of a fan is expressed in terms of air horsepower (*a.hp.*) and represents work done by the fan.

$$A.hp. = \frac{W_a H}{33{,}000}. \tag{105}$$

W_a = weight of air handled per minute, lb.
H = total head against which the fan works, ft. of air.

In terms of the total pressure difference created by the fan,

$$a.hp. = \frac{Q \times 60 \times h_T \times D}{12 \times 33{,}000}. \tag{106}$$

Q = volume of air handled, cu. ft. per sec.
h_T = total pressure difference created by the fan, in. of water.
D = density of one cubic foot of water at the temperature of the gage fluid, lb.

Example.—A fan discharges 24,000 cu. ft. of air per min. through its outlet of 12.6 sq. ft. area and maintains a static pressure of 5 in. of water. Air temperature, 70 deg. F. Barometer, 29.92 in. of mercury. Find the air horsepower developed.

Solution.—

$$V_a = \frac{24{,}000}{12.6 \times 60} = 31.8 \text{ ft. per sec.}$$

$$h_{v_a} = \frac{12V_a^2 d}{2gD} = \frac{31.8^2 \times 12 \times 0.075}{2 \times 32.17 \times 62.3} = 0.226 \text{ in. of water.}$$

$$h_T = h_s + h_{v_a} = 5. + 0.226 = 5.226 \text{ in. of water.}$$

$$A.hp. = \frac{24{,}000 \times 5.226 \times 62.3}{12 \times 33{,}000} = 19.8.$$

162. Mechanical Efficiency of Fans.—The ratio of the air horsepower output to the driving power required at the fan shaft expresses the mechanical efficiency of the fan.

$$Fan\ mechanical\ efficiency,\ e_m = \frac{air\ horsepower}{horsepower\ input}. \qquad (107)$$

The mechanical efficiencies of steel-plate fans range from 40 to 60 per cent, and those of multibladed fans from 50 to 80 per cent. Each fan has a condition of maximum efficiency when working against any static pressure. Fan builders usually signify, by bold-face type in their

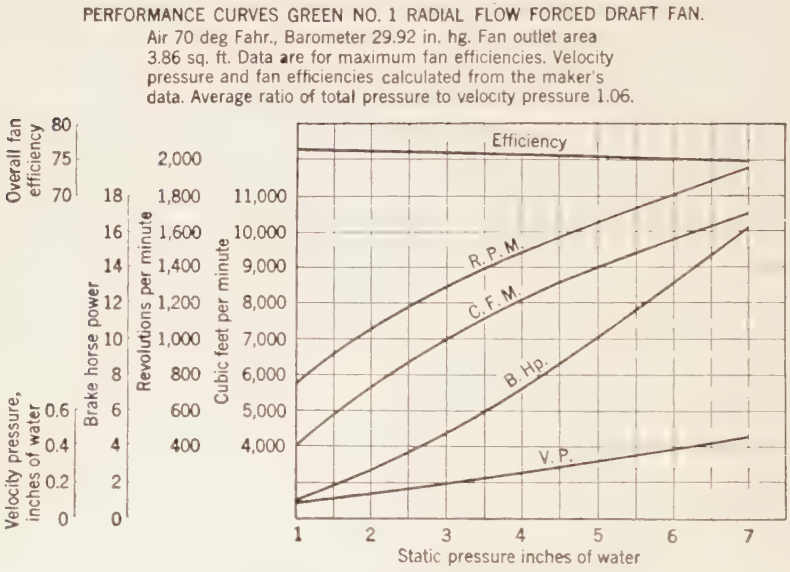

PERFORMANCE CURVES GREEN NO. 1 RADIAL FLOW FORCED DRAFT FAN.
Air 70 deg Fahr., Barometer 29.92 in. hg. Fan outlet area 3.86 sq. ft. Data are for maximum fan efficiencies. Velocity pressure and fan efficiencies calculated from the maker's data. Average ratio of total pressure to velocity pressure 1.06.
Overall fan efficiency
Brake horse power
Revolutions per minute
Cubic feet per minute
Velocity pressure, inches of water
Efficiency
R.P.M.
C.F.M.
B.Hp.
V.P.
Static pressure inches of water

FIG. 134.—Fan performance curves.

capacity tables, the speed and brake horsepower necessary to give the maximum efficiency at various static pressures.

163. Fan Performance.—Fan characteristics and performance are dependent upon the design and conditions of operation, that is, speed, temperature of the gas handled, static pressure, and the velocity pressure maintained. Table XVII gives the performance data of multiblade fans suitable for forced draft. These numerical quantities were selected from the builder's data as those representing a condition of maximum efficiency at the various static pressures. The individual performances of a single fan selected from the data of Table XVII are shown graphically by the curves of Fig. 134.

TABLE XVII

CAPACITIES, SPEEDS, AND HORSEPOWERS, GREEN RADIAL-FLOW DOUBLE-INLET, FORCED-DRAFT FANS

(Green Fuel Economizer Co.)

Air temperature, 70 deg. F. Barometer, 29.92 in. of mercury
Table values are for maximum efficiency

Fan No.	Wheel Dia., In.	Per-formance	Static Pressure at Fan Discharge, Inches of Water						
			1	2	3	4	5	6	7
0	20½	C.f.m.	2,800	4,000	5,200	6,000	6,400	7,200	7,600
		R.p.m.	860	1,216	1,505	1,738	1,927	2,122	2,280
		B.hp.	0.62	1.77	3.49	5.36	7.15	9.65	11.85
1	24	C.f.m.	4,000	5,500	7,000	8,000	9,000	9,500	10,500
		R.p.m.	741	1,043	1,284	1,481	1,658	1,805	1,957
		B.hp.	0.87	2.47	4.69	7.15	10.01	12.8	16.4
2	27¾	C.f.m.	5,600	7,700	9,100	10,500	11,900	12,600	14,000
		R.p.m.	648	913	1,110	1,280	1,435	1,563	1,696
		B.hp.	1.22	3.41	6.11	9.39	13.4	17.0	21.9
3	32¼	C.f.m.	7,200	9,900	12,600	14,400	16,200	17,100	18,900
		R.p.m.	550	776	955	1,102	1,232	1,342	1,456
		B.hp.	1.57	4.41	8.46	12.9	18.1	22.9	29.5
4	37¼	C.f.m.	10,000	13,750	16,250	18,750	21,250	23,750	25,000
		R.p.m.	478	671	816	944	1,055	1,161	1,248
		B.hp.	2.18	6.13	10.9	16.8	23.8	31.8	39.0
5	43	C.f.m.	13,500	18,000	22,500	25,500	28,500	31,500	33,000
		R.p.m.	416	583	714	821	917	1,007	1,083
		B.hp.	2.97	8.06	15.1	22.8	31.9	42.3	51.6
6	50	C.f.m.	18,000	24,000	30,000	36,000	38,000	42,000	46,000
		R.p.m.	354	495	609	707	782	859	930
		B.hp.	3.95	10.7	20.1	32.3	42.5	56.3	71.9
7	58	C.f.m.	22,000	34,000	40,000	46,000	52,000	58,000	61,000
		R.p.m.	301	433	525	606	678	746	803
		B.hp.	4.86	15.2	26.9	41.3	58.0	77.8	95.9
8	67¼	C.f.m.	30,000	46,000	54,000	62,000	70,000	78,000	82,000
		R.p.m.	260	372	452	521	583	641	689
		B.h.p.	6.57	20.5	36.1	55.6	78.1	104	128
9	78	C.f.m.	45,000	60,000	75,000	85,000	95,000	105,000	110,000
		R.p.m.	228	319	391	451	504	553	594
		B.hp.	10.2	26.8	50.2	76.0	107	141	172
10	90½	C.f.m.	57,500	80,000	102,500	117,500	125,000	140,000	155,000
		R.p.m.	194	274	337	389	431	474	515
		B.hp.	12.7	35.5	68.5	105	140	187	242

Fan capacity varies directly as the speed ratio; the static pressure developed, as the square of the speed ratio; and the power required to drive the fan, as the cube of the speed ratio. Thus, doubling the speed of a fan doubles the volume of its discharge, produces a static pressure four times as great, and requires eight times as much power to drive the fan. An examination of the data of Table XVII will verify the foregoing statement.

Example.—A No. 1 fan at 741 r.p.m. delivers 4000 c.f.m. at 1-in. static pressure and requires 0.87 b.hp. to drive it. When the speed is doubled, or 1481 r.p.m., the discharge is 8000 c.f.m., the static pressure 4 in. of water, and the b.hp. 7.15 or 8.2 times as much as at the 741-r.p.m. speed.

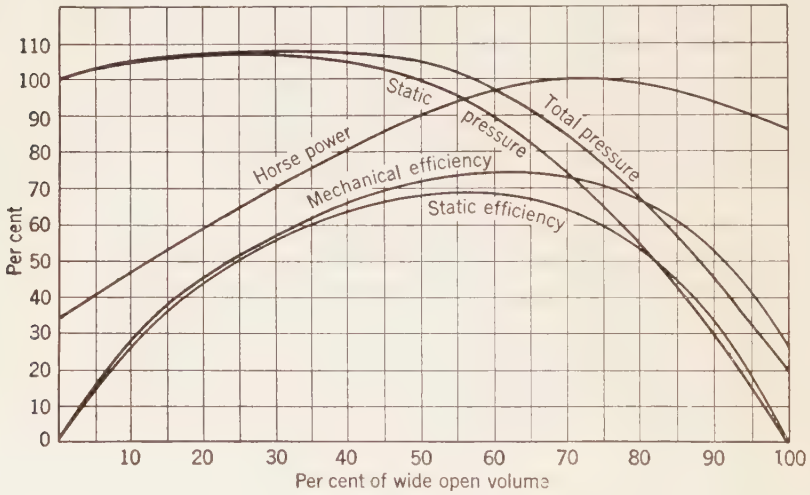


Fig. 135.—Characteristic curves, type RB Green radial-flow forced-draft fans.

Fans may be operated to give variable capacities with varying efficiencies at any static pressure. With the aid of a set of curves as shown in Fig. 135, the performances of any of the fans listed in Table XVII may be ascertained when the operation is at various capacities. The static efficiency is the mechanical efficiency multiplied by the ratio of the static to the total pressure.

PROBLEMS

1. Find the probable draft at the base of a chimney 175 ft. high when the outside temperature is 90 deg. F. and the average temperature of the chimney gases is 350 deg. F. Draft gage temperature 95 deg. F.

2. What force of draft may be expected from a chimney 175 ft. high if the outside temperature is −10 deg. F. and the average temperature of the chimney gases is 350 deg. F.? Boiler room temperature 65 deg. F.

3. A chimney is to handle gases at an average temperature of 400 deg. F. and is to give an actual draft of 0.9 in. of water at its base when the outside temperature is 100 deg. F. What must be the height of the chimney?

4. A boiler plant has the following actual draft losses in inches of water: through fuel bed, 0.25; boiler passes, 0.32; straight breeching, 0.15; velocity of flow, 0.04; and two bends in breeching, 0.05 each. Average temperature of chimney gases, 350 deg. F. Outside air temperature, 80 deg. F. Find the height in feet of the chimney required.

5. What actual velocity of flow of gases may be expected in a chimney 200 ft. high when the average gas temperature is 500 deg. F. and the outside air temperature 98 deg. F.? Velocity coefficient, 0.40.

6. Design a chimney for a boiler plant using the following data: actual draft required, 0.85 in. of water; coal used per hour, 25,000 lb.; dry flue gases per pound of coal fired, 14; temperature of the flue gases in the breeching, 650 deg. F.; average temperature of the chimney gases, 450 deg. F.; outside air temperature, 95 deg. F.

7. What is the boiler horsepower capacity of a chimney 225 ft. high and 16 ft. inside diameter?

8. Air flowing in a duct had a total pressure of 1.15 in. of water and a velocity pressure of 0.17 in. of water when the air temperature was 185 deg. F. and the barometric pressure was 28.75 in. of mercury. Find the static pressure and the average velocity of flow in feet per second.

9. Air flowed in a duct 3 ft. in diameter at a velocity of 30 ft. per sec. and a static pressure of 0.65 in. of water. The air temperature was 60 deg. F. and the barometric pressure 29.7 in. of mercury. Find the total pressure of the air. How many cubic feet of air flowed per minute?

10. A fan discharged 6000 cu. ft. of air per min. through a duct 4 sq. ft. in cross-sectional area against a static pressure of 0.5 in. of water. The air temperature was 70 deg. F. and the barometric pressure was 29.0 in. of mercury. Find the air horsepower developed. What brake horsepower was necessary to drive the fan if the fan efficiency was 62 per cent?

11. Calculate the mechanical efficiency of a No. 9 Green radial-flow fan operating with 7 in. of water static pressure and discharging the air into a duct 75 in. by 80 in. Air temperature, 70 deg. F.; barometer, 29.92 in. of mercury.

12. Find in inches of water the total pressure difference produced by a fan discharging 6280 cu. ft. of gases per min. with a velocity of 2000 ft. per min. and a static pressure of 0.20 in. of water. The diameter of the suction pipe was 30 in., and the static pressure in the suction duct was −0.63 in. of water. Gas temperature, 250 deg. F. Barometric pressure, 29.92 in. of mercury.

13. A fan discharged air into a duct 4 ft. in diameter under a static pressure of 1.02 in. of water and a velocity of 50 ft. per sec. Air temperature was 85 deg. F., and the barometer was 29.92 in. of mercury. In the inlet duct to the fan the static pressure was −0.95 in. of water and the velocity pressure was 0.50 in. of water. Find the air horsepower developed. What brake horsepower was necessary to drive the fan if the mechanical efficiency was 70 per cent?

14. Find the static efficiency of a fan discharging 10,500 cu. ft. of air per min. against a discharge static pressure of 4 in. of water; air velocity, 1800 ft. per min.; brake horsepower to drive the fan, 9.4. Air temperature, 70 deg. F. Barometer, 29.92 in. of mercury.

15. Calculate, from the data of Table XVII, the static efficiency of a No. 10 fan operating with a static pressure of 6 in. of water. Fan outlet 87 in. by 93 in.

CHAPTER XI

THE STEAM ENGINE

164. Application.—As a source of power the steam engine has the following characteristics as a prime mover: (1) ability to start under load; (2) nearly constant, slow, or comparatively slow rotative speeds; (3) variable rotative speeds; and (4) direction of rotation may be reversed. Not every steam engine can satisfy all the foregoing items, but by proper design all of those mentioned may be incorporated in a single machine as in the modern steam locomotive.

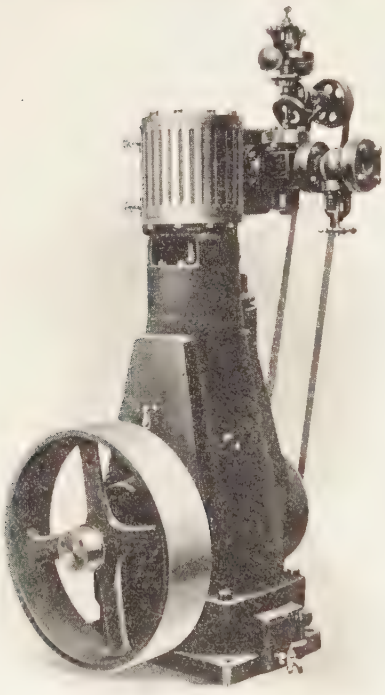


FIG. 136.—Troy vertical steam engine.

The steam engine is not suited to extremely high rotative speeds and in condensing practice is limited to operation with a maximum vacuum, referred to a 30-in. barometer, of about 25 to 26 in. of mercury in the exhaust piping. In general, steam-engine units of good design up to 2000 hp. in size are as economical as or more economical than steam turbines of similar size operating with comparable conditions. For the production of large quantities of electrical energy the steam engine has been generally superseded by the steam turbine. High rotative speeds are desirable in alternating-current generation. The steam turbine can develop a large amount of power in a machine occupying a very much smaller floor space than would be required for steam engines of equal capacity. The turbine can operate with high initial steam pressures and temperatures as well as with vacua, referred to a 30-in. barometer, of 28 to 29 in. of mercury in the exhaust connections. The latter condi-

tions secure for the turbine an economy of operation that is not equaled by the steam engine.

Steam engines are largely used for driving pumps, fans, blowers, line shafts, rolling-mill machinery, hoisting equipment, electrical generators of limited sizes, refrigerating machinery, and air compressors.

165. The Simple Engine.—The simple engine generally has only a single steam cylinder, although twin cylinders may be used. In the simple engine, partially expanded steam is not discharged from one cylin-

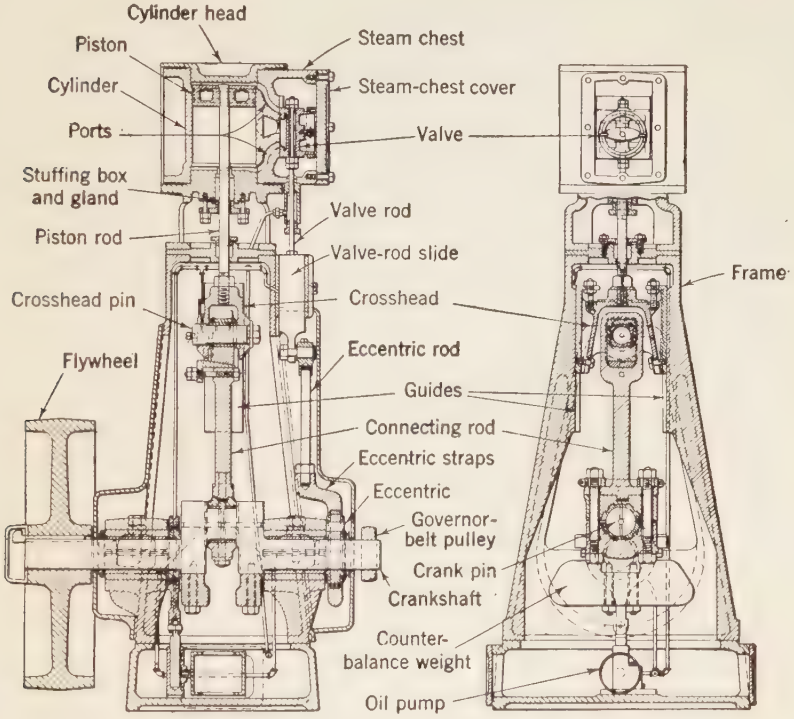


Fig. 137.—Sectional-elevation of a simple vertical engine.

der to another for further expansion of the steam. Figure 136 illustrates a small vertical steam engine with a throttling governor and a D-slide-valve. Such engines are suitable for the development of small amounts of power and are used for driving fans, blowers, stokers, small electric generators, etc. The essential parts of the engine (Fig. 136) are illustrated and designated by Fig. 137. The engine is double acting because a power cycle is produced in each end of the cylinder during one revolution of the crankshaft.

Steam from the supply line is admitted by the throttle valve and passes through the governor to the steam chest. The D-slide-valve alternately admits steam to and releases the steam from each end of the cylinder. The valve has a reciprocating motion produced by the gear composed of the eccentric, the eccentric straps, the eccentric rod, the valve slide, and the valve rod. As shown by Fig. 138, the valve is in a position to admit steam to the upper end of the cylinder and to allow the escape of used steam from the lower end. The piston is moved by the pressure of the steam admitted to the cylinder. At some point of the piston stroke the valve completely closes the supply port and the flow of steam from the steam chest is cut off. After cut-off has taken place the pressure within the cylinder decreases, owing to expansion of the steam. Energy stored in the heavy flywheel during the period of steam admission is used to maintain the engine speed. Near the end of the piston stroke the valve moves to uncover the steam port of the cylinder end under consideration. When this movement takes place the expanded steam is released and flows to the exhaust passage underneath the valve as shown for the lower (crank) end of the cylinder, Fig. 138. The exhaust of expanded steam underneath the valve occurs while live steam is being admitted to the opposite (head) end of the cylinder and continues until near the end of the return stroke of the piston. When the valve has moved to completely close both ports of the cylinder, some expanded steam is entrapped ahead of the moving piston and is compressed with a rise in pressure. This compression acts as a cushion for the returning piston and continues until the valve has moved to admit live steam to that end of the cylinder. One such complete cycle takes place in each end of the cylinder per revolution.

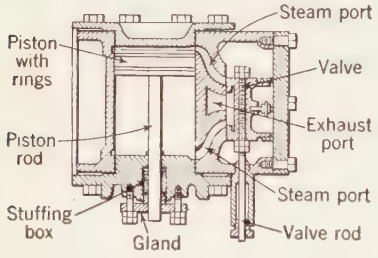


Fig. 138.—Section of a simple steam engine cylinder.

The reciprocating motion imparted to the piston by the alternate admission, expansion, and release of steam in each end of the cylinder is transmitted by the piston rod to the crosshead. This part moves back and forth between the crosshead guides and delivers its motion to one end of the connecting rod. The connecting rod is fitted at each end with a pin joint and converts the reciprocating motion of the crosshead to rotary motion at the crankshaft. The pin passing through the crosshead and one end of the connecting rod is known as the crosshead or wrist pin. The pin at the crankshaft end of the connecting rod is called the crank pin.

166. Engine Classifications.—Steam engines are divided into a number of types, depending upon their design, construction, and method of operation. The common classifications used are:

(1) Position of the frame and cylinder { Horizontal / Vertical

(2) Valve gear { Slide-valve { D / Balanced / Multiported / Piston ; Corliss { Releasing / Non-releasing ; Poppet

(3) Number of cylinders used for steam expansion { Single cylinder, simple / Two cylinders, compound / Three cylinders, triple expansion / Four cylinders, quadruple expansion

(4) Rotative speed { Low / Medium / High

(5) Ratio of stroke to diameter { Long / Short

(6) Exhaust disposal { Condensing / Non-condensing

167. Engine Nomenclature.—In steam-engine parlance the following terms are used: *Head end* of the cylinder is farther from, and the *crank-end* is nearer to, the crankshaft. *Dead center* is the position at which reversal of piston motion occurs at either end of the cylinder. In the dead-center position the longitudinal center lines of the crank, the connecting rod, and the piston rod are all in perfect alignment. An engine has two dead centers, that is, head-end and crank-end. *Stroke* is the linear distance traveled by the piston from one dead-center position to the other. The forward stroke is made between the head-end and the crank-end dead centers, and the return stroke is completed with motion in the opposite direction. Long-stroke engines have a piston travel greater than, whereas short-stroke engines have piston travels equal to or less than, the cylinder diameter. *Piston displacement* is the volume swept through by the piston during a stroke and is equal to the product of the piston area and the stroke length.

Clearance is either mechanical or volumetric. Mechanical clearance is the smallest linear distance between the cylinder head and the piston

at the dead-center position. Volumetric clearance is generally used and is the volume of the space between the piston at dead center and the cylinder head, together with the volume of the steam port at that cylinder end. Clearance thus defined is expressed as a percentage of the piston displacement. Clearance percentages are greater for high-speed, short-stroke engines than for long-stroke, slow-speed engines. *Rotative speeds* greater than 250 r.p.m. are classed *high*, between 150 to 250 r.p.m. *medium*, and less than 150 r.p.m. *slow*.

168. Steam Cycles.—The Carnot and the Rankine cycles of the working fluid are the usual standards by which steam-engine performance may be judged. Theoretically the Carnot cycle, discussed in Arts. 19 and 20, is the most efficient for engines working between given temperature limits. However, the cycle is impossible in the commercial

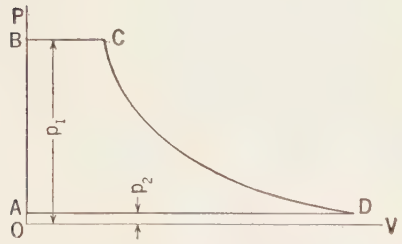


Fig. 139.—Rankine cycle with complete expansion.

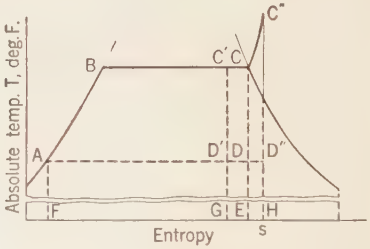


Fig. 140.—Temperature-entropy diagram for the Rankine cycle with complete expansion.

steam engine as the water cannot be vaporized and condensed in the engine cylinder.

The Rankine cycle.—The performance of the actual steam engine partially fulfills the conditions of the ideal standard set up as the Rankine cycle. This ideal performance standard is based on the premises that the engine operates without clearance, compression of the steam, or pressure losses, and that the steam is expanded adiabatically.

The Rankine cycle with complete expansion of the steam is illustrated by the pressure-volume diagram of Fig. 139 and the temperature-entropy diagram of Fig. 140. The cycle really represents the operation of the plant in which the liquid or feedwater is heated in a boiler along the line AB with a rise in temperature and pressure; vaporization of the steam in the boiler follows along the line BC at constant pressure; the steam formed is then expanded adiabatically within the engine cylinder from C to the exhaust pressure at D; and finally exhaust at constant pressure and temperature takes place along DA. No matter what disposition is made of the steam, either by condensation or exhaust to the atmosphere, the temperature of the feedwater supplied to the

boiler at A is assumed to be that corresponding to the temperature of the exhaust steam along line DA. On the T–s diagram, area $ABCD$ represents the heat utilized or the work done by dry saturated steam; area $ABC'D'$, by wet steam of quality x; and area $ABC''D''$, that done by superheated steam. The areas $ADEF$, $AD'GF$, and $AD''HF$ represent, respectively, the heat rejected. The total heat of the steam in each case is shown by $FABCE$, $FABC'G$, and $FABC''H$.

The efficiency of the Rankine cycle with complete expansion of the steam is the ratio of the heat utilized to the heat available in the cycle above the temperature at the exhaust pressure.

$$e_R = \frac{h_1 - h_2}{h_1 - h_{f,2}}, \tag{108}$$

where e_R = Rankine-cycle efficiency.
h_1 = total initial heat of dry, wet, or superheated steam at C, C' or C'', B.t.u. per lb.
h_2 = total heat of steam at D, D', or D'' after adiabatic expansion, B.t.u. per lb.
$h_{f,2}$ = heat of the liquid at exhaust pressure DA, B.t.u. per lb.

Steam engines built to expand steam completely to the pressure at D require a larger cylinder volume than is necessary for incomplete expansion of the steam. The work obtained by steam expansion beyond a certain point is small; therefore the commercial practice is to expand the steam to some pressure L above that of the exhaust (Fig. 141) and then release it from the cylinder with a rapid pressure drop LM to that of the exhaust line. The small loss in power in the Rankine cycle with incomplete expansion illustrated by the sectioned portion, Fig. 141, is more than offset by the additional initial and operating costs of an engine designed to operate with complete expansion of the steam.

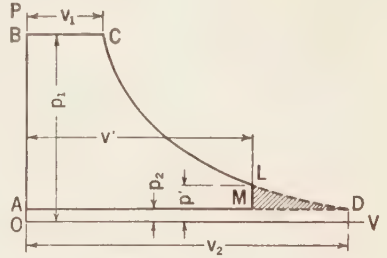


Fig. 141.—Rankine cycle with incomplete expansion.

The heat utilized in the incomplete expansion cycle is not the same as that used in the complete expansion cycle. As the steam volume at release, with incomplete expansion, is v' instead of v_2, owing to the limitation of the cylinder dimensions, the steam volume v' remains constant as steam escapes from the cylinder, at release, and the pressure drops from p' to p_2. Consequently, the work done by the escaping steam in dropping from the release pressure p' to the exhaust pressure

p_2 is equal to $[144(p' - p_2)v'] \div 778$. The heat of the steam after adiabatic expansion from pressure, p_1 to p' is h' B.t.u. per lb.; therefore the efficiency of the cycle with incomplete expansion is

$$e_R = \frac{h_1 - h' + 0.1851(p' - p_2)v'}{(h_1 - h_{f,2})}. \tag{109}$$

169. Theoretical Power.—The engine described in Art. 165 is not the most efficient thermally, but is one of the simplest types of engines built. The discussion of theoretical power developed by such a heat motor is applicable to all classes of engines. Assumptions are made that the conditions of incomplete expansion of the steam prevail and that the operation is without clearance, compression, or reduction of the steam pressure during admission of steam to the cylinder. It is presumed that the valves function instantly, and that the equation of the expansion curve is that of a rectangular hyperbola, $PV = C$. Under such conditions the theoretical indicator card or pressure-volume diagram for one end of the cylinder is as illustrated by Fig. 142.

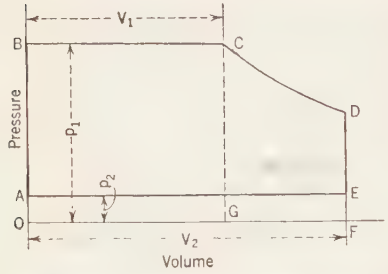


FIG. 142. Theoretical pressure-volume diagram.

Steam admitted to the cylinder at A enters at a constant pressure p_1 pounds per square inch absolute until the point of instant cut-off occurs at C. At this time a volume of v_1 cubic feet of steam has entered the cylinder. Expansion of the steam takes place along the line CD with an increase of volume to v_2. Rapid release at D allows the steam pressure to drop to that of exhaust, p_2 pounds per square inch absolute. Exhaust occurs at constant pressure p_2 and continues until live steam is again admitted to the cylinder at A.

The total area of the pressure-volume diagram $OBCDF$ represents the work done by the steam. The part $OAEF$ depicts the work necessary to force out the steam during exhaust. Consequently, the net work done is represented by the area $ABCDE$.

$$ABCDE = OBCG + GCDF - OAEF$$

$$= p_1 v_1 + \int_{v_1}^{v_2} \frac{dv}{v} - p_2 v_2.$$

The work indicated by the net area can be put into an equation of a

different form if the expansion ratio $\frac{v_2}{v_1} = r$ is considered, together with the fact that the equation of curve CD is

$$p_1v_1 = p_2v_2 = pv = c,$$

hence,

$$p = \frac{p_1v_1}{v},$$

$$ABCDE = p_1v_1 + p_1v_1\int_{v_1}^{v_2}\frac{dv}{v} - p_2v_2$$

$$= p_1v_1\left(1 + \int_{v_1}^{v_2}\frac{dv}{v}\right) - p_2v_2.$$

When the integration is made and r is substituted for $\frac{v_2}{v_1}$, the work may be expressed as

$$ABCDE = p_1v_1(1 + \log_e r) - p_2v_2. \qquad (110)$$

Equation 110 expresses the work done within a single end of the cylinder per revolution of the crankshaft. The mean net steam pressure per stroke pushing the piston forward is the *mean effective pressure* (*m.e.p.*). This pressure is calculated by dividing the work area of the card by the extreme length of the card. Thus, if equation 110 is divided by the diagram length v_2, the mean effective pressure is

$$p_m = \frac{p_1v_1(1 + \log_e r) - p_2v_2}{v_2} = p_1\left(\frac{1 + \log_e r}{r}\right) - p_2. \qquad (111)$$

The mean effective pressure thus determined is a theoretical value, and the actual m.e.p. is always smaller on account of certain thermal and mechanical losses which will be discussed later.

The theoretical indicated horsepower developed per end of the cylinder is expressed as

$$\text{i.hp.} = \frac{p_mlAN}{33{,}000}. \qquad (112)$$

p_m = theoretical mean effective pressure, lb. per sq. in.
l = length of stroke, ft.
A = piston area, sq. in.
N = number of revolutions per minute.

The total average pressure in pounds exerted upon the piston is $p_m \times A$, and as this force acts through a distance of l feet, the work in foot-

pounds done per stroke is $p_m Al$. A working stroke is completed for each cylinder end during a revolution of the crankshaft and flywheel, and the work done per minute per cylinder end is $p_m lAN$.

Example.—A 7-in. by 10-in. double-acting steam engine is to operate at 200 r.p.m. with the initial steam pressure at 150 lb. per sq. in. abs. and the exhaust pressure at 15 lb. per sq. in. abs. Cut-off is to be at $\frac{5}{8}$ stroke. Clearance and the piston rod are to be neglected. Find the theoretical horsepower of the engine.

Solution.—Apparent number of steam expansions $= r = \frac{8}{5} = 1.6$

$$p_m = 150\frac{(1 + \log_e 1.60)}{1.60} - 15 = 122.8 \text{ lb. per sq. in.}$$

$$\text{Theoretical i.hp.} = \frac{2 \times 122.8 \times \frac{10}{12} \times 38.48 \times 200}{33{,}000} = 47.8.$$

170. The Actual Steam-engine Cycle.—Mechanical and thermal limitations in the construction and operation of steam engines do not

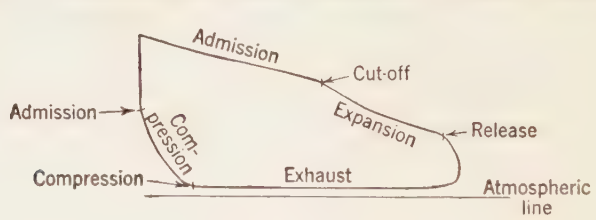


Fig. 143.—Actual pressure-volume diagram.

permit the development of the theoretical power of any engine. Engine clearance volume cannot be eliminated, and some compression of the steam entrapped in the clearance is desirable to cushion the reciprocating parts as reversal of the direction of motion occurs at the end of each stroke. This compression of steam lessens the net useful power developed within the cylinder. Losses in steam pressure often occur as the steam flows through the ports to the cylinder and the engine valves do not function instantly at cut-off and release. The reduction of steam pressure caused by friction losses in the ports and the throttling action of valves in opening or closing is known as *wiredrawing*. Also thermal losses occur within the cylinder.

As a result of the foregoing losses, the actual pressure-volume diagram obtained from an engine shows less power developed than the theoretical diagram for the same operating conditions. By the use of an engine indicator a diagram may be taken to show the action of the steam within the cylinder. A sample diagram from a simple slide-valve engine is illustrated by Fig. 143. The beginning and end of the various

events of the cycle are marked. Indicator diagrams from an engine more economical in the use of steam are shown in Art. 221; see Fig. 181. By means of measurements and computations explained later, the mean effective pressure of the actual diagram can be obtained. The ratio of the mean effective pressure of the actual diagram to the mean effective pressure of the theoretical diagram is the *diagram factor*. This ratio or factor for small simple throttling governor slide-valve engines ranges from 0.58 to 0.77. Engines of better design which expand the

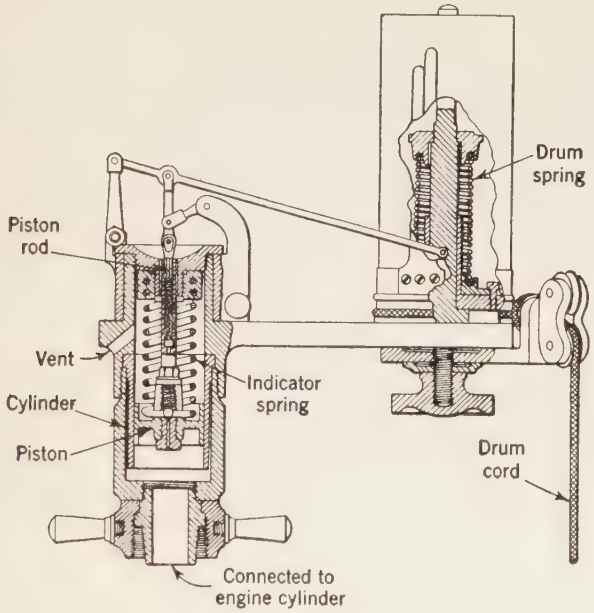


Fig. 144.—Crosby inside-spring indicator.

steam a greater number of times have diagram factors ranging from 0.80 to 0.90.

171. Engine Indicators.—Instruments for drawing actual pressure-volume diagrams of the steam cycle are of two general types, inside and outside spring indicators. As the names imply, the springs resisting the movement of the indicator piston are either within the indicator cylinder, Fig. 144, or exterior to it, Fig. 145. In principle of operation, the indicators are the same, although the pencil and piston mechanisms of the two differ somewhat.

Either indicator is attached to suitable piping leading to the clearance spaces at the ends of the engine cylinder. When a cock in the piping connections is open, the indicator piston is subjected to the varying pressures within the engine cylinder. Pressure exerted upon the indi-

cator piston overcomes the resistance of the indicator spring in varying amounts and causes the piston rod to move. The motion of the piston rod is transmitted through a carefully designed linkage system to give movement to a lever carrying a pencil point or stylus which moves at all times parallel to the line of motion of the indicator piston. Indicator springs are made and calibrated to allow a vertical movement of the pencil point through a distance of one inch when a given pressure in pounds per square inch is applied to the indicator piston. This rating in pounds per square inch for a pencil movement of one inch is the indicator *spring scale.*

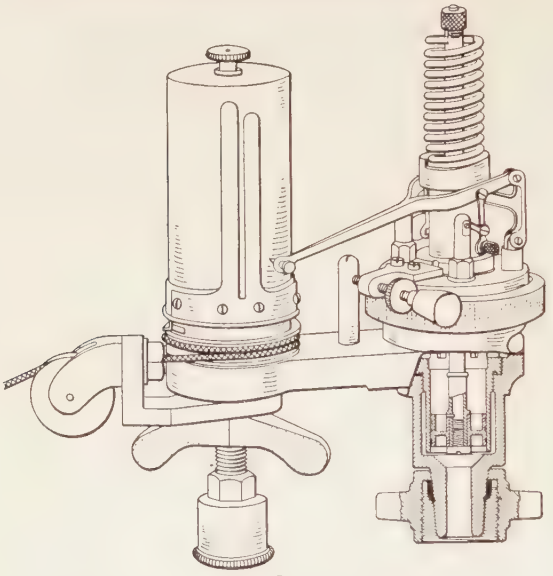


Fig. 145.—Maihak standard indicator.

The pencil or stylus is used to make a record upon a sheet of paper placed about the indicator drum. The drum is rotated forward and backward by a cord wound around the lower part and the action of a coil spring within the drum. The indicator cord should run over a properly located guide pulley and is attached to some part of a reducing motion driven preferably by the engine crosshead.

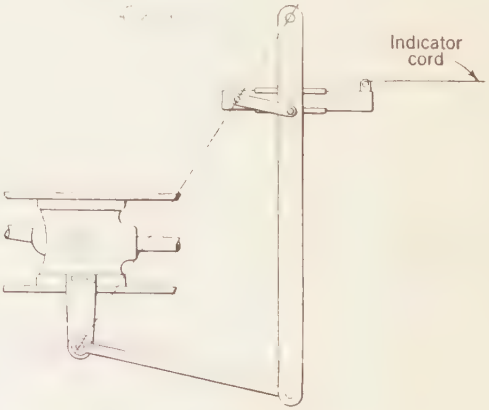


Fig. 146.—Indicator reducing motion.

The function of the reducing motion is to reproduce exactly to a smaller scale the reciprocating motion of the engine piston. Unless the reduction of motion is exact at all points of the piston travel, the pres-

sure-volume diagram produced by the indicator will be inaccurate. Such a condition leads to an erroneous representation of the pressure and volume relations existing within the engine cylinder. A lever type of reducing motion that gives satisfactory results is illustrated by Fig. 146.

The pressure-volume diagram obtained by the use of an indicator with an exact reducing motion may be used for the following purposes:

(1) A check of the steam distribution discloses the points at which the steam valve or valves open and close. From this information a study of the adjustments of the valve and the valve gear may be made.
(2) Determination of the average or mean effective pressure existing within the cylinder during a cycle.
(3) Analyses of the actual action of a steam engine.

172. Mean Effective Pressure.—The work areas of indicator diagrams are usually measured by mechanical integrating devices known

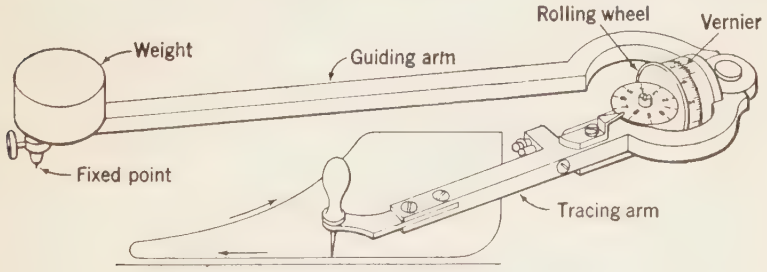


FIG. 147.—Fixed-arm polar planimeter.

as polar planimeters. Such a mechanism, Fig. 147, when properly constructed, adjusted, and handled, will rapidly give area measurements of sufficient accuracy. When the area of a diagram is known the mean height or mean ordinate O may be determined by dividing the diagram area in square inches by the extreme length, in inches, of the enclosed area. The mean ordinate in inches is the height of a rectangle whose area is equal to that of the diagram and whose length is equal to the extreme length of the diagram.

The mean effective pressure p_m within the engine cylinder is equal to the product of the mean ordinate O of the indicator diagram, and the scale S of the spring used when the diagram was taken.

$$p_m = O \times S \tag{113}$$

173. Indicated Horsepower.—The power produced within the ends of a double-acting steam cylinder is expressed as

$$\text{i.hp., head end} = \frac{p_m lAN}{33{,}000}, \tag{114}$$

$$\text{i.hp., crank end} = \frac{p'_m lA'N}{33{,}000}, \tag{115}$$

in which p_m = head-end m.e.p., lb. per sq. in.
p'_m = crank-end m.e.p., lb. per sq. in.
l = length of engine stroke, ft.
A = net area of piston, head end, sq. in.
A' = net area of piston, crank end, sq. in.
N = revolutions per minute.

The quantities $\frac{lA}{33{,}000}$ and $\frac{lA'}{33{,}000}$ of equations 114 and 115 are always fixed for an engine. The ratios when expressed numerically for the cylinder ends are known respectively as the head- or crank-end *engine constants*. The use of the ratios facilitates the rapid calculation of the power developed when the engine r.p.m. or m.e.p. varies. When a piston rod is attached to the piston in either end of the cylinder, its cross-sectional area must be deducted from the gross piston area to give the net effective piston area.

$$\text{Total i.hp.} = \frac{p_m lAN}{33{,}000} + \frac{p'_m lA'N}{33{,}000}. \tag{116}$$

When engine cylinders are single-acting, power is not developed in the crank end and the indicated horsepower is that of the head ends alone. The total indicated horsepower of engines having more than one cylinder is the sum of the indicated horsepowers of all the cylinders.

174. Brake Horsepower.—The actual power output delivered by the crankshaft or at the rim of the flywheel of an engine is the brake horsepower. On account of friction and windage losses occasioned by the moving parts of the engine, the brake horsepower delivered is always less than the indicated horsepower developed.

The brake horsepower of comparatively small engines is determined by the use of a Prony brake, Fig. 148. The brake consists of a flexible band or strap carrying a number of wooden blocks, which encircle the engine flywheel or a brake pulley mounted upon the engine shaft. The wooden blocks are in contact with the surface of the wheel, and the strap or band may be tightened or loosened upon the brake wheel by means of a hand adjustment. The brake band is rigidly attached to the two

lever arms which meet at a common knife-edge fulcrum C. The horizontal distance from C to the center of the engine shaft is the length r of the brake arm. The free end of the brake arm at C should rest upon a pedestal set upon the platform of weighing scales. This pedestal should be of sufficient height to maintain C at the same elevation as that of the engine shaft center line. The Prony brake is a friction device, and the power developed at the rim of the brake wheel is dissipated as heat. Therefore, provision must be made for cooling the rim of the brake pulley. Water is commonly used as a cooling medium and is held within the pulley rim by internal rim flanges.

The power absorbed by the brake of Fig. 148 is calculated in the following manner. R is equal to the resistance in pounds offered to motion

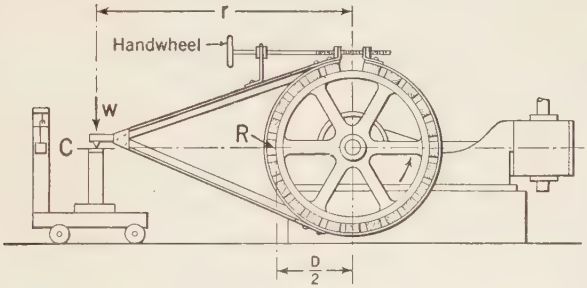


FIG. 148.—Prony brake.

at the rim of a pulley D feet in diameter. The torque developed at the rim is equal to $R \times \frac{D}{2}$ foot-pounds, and this is also equal to the torque developed at C, or $w \times r$; w being the net pressure applied to the brake pedestal due to the action of the brake and r being the effective brake arm in feet.

$$R \times \frac{D}{2} = wr \quad \text{or} \quad RD = 2wr.$$

The force R is applied to the periphery $\pi \times D$ of the wheel which makes N revolutions per minute. The work output per minute is πDRN or $2\pi rwN$ foot-pounds, and

$$\text{b.hp.} = \frac{2\pi rwN}{33{,}000}. \tag{117}$$

In equation 117 the quantity $\frac{2\pi r}{33{,}000}$ is fixed for a brake and the ratio is the *brake constant.*

Example.—A steam engine equipped with a Prony brake having an arm $5\frac{1}{4}$ ft. long ran at 200 r.p.m. The net load at the end of the arm was 125 lb. Find the brake horsepower.

Solution.—

$$\text{b.hp.} = \frac{2 \times 3.1416 \times 5\frac{1}{4} \times 125 \times 200}{33{,}000} = 25.$$

175. Friction Horsepower.—The loss of mechanical power within an engine is known as friction horsepower. This quantity is equal to the difference of the total i.hp. and the b.hp. Friction horsepower varies with the percentage of rating developed, size, type, rotative speed, and mechanical construction of an engine. In general, the percentage of friction horsepower is greater in multicylinder engines than in simple engines.

176. Mechanical Efficiency.—The ratio of the engine power output to the power developed within the cylinder is the mechanical efficiency of an engine.

$$\text{Mechanical efficiency, } e_m = \frac{\text{b.hp.}}{\text{i.hp.}}. \tag{118}$$

The mechanical efficiencies of steam engines at normal loads range from 85 to 95 per cent, depending upon the design and workmanship of the engine. Poorly constructed engines may have the lower figure and high-grade machines may give the latter value. The mechanical efficiency of an engine varies with the load upon the engine, increasing, with the load, from light loads to a maximum, and then decreasing with a further increase of engine load.

PROBLEMS

1. An 8-in. by 18-in. steam engine has a piston rod $1\frac{7}{8}$ in. in diameter. Find the piston displacement in cubic feet per stroke for each end of the cylinder. Find the piston speed in feet per minute when the engine runs at 100 r.p.m.

2. Calculate the piston displacement in cubic feet for each cylinder end of a 10-in. by 12-in. steam engine whose piston rod is $1\frac{1}{2}$ in. in diameter. Find the piston speed in feet per minute when the engine speed is 240 r.p.m.

3. A 10-in. by 12-in. steam engine with a piston rod $1\frac{1}{2}$ in. in diameter required 3.12 lb. of water to fill the head-end clearance space and 3.38 lb. of water to fill the crank-end clearance space. The water temperature was 70 deg. F. Find the clearance percentages.

4. Find the clearance volumes in cubic feet of a 7-in. by 10-in. steam-engine cylinder fitted with a piston rod $1\frac{3}{16}$ in. in diameter, if the clearance percentages are: head end, 15.8; and crank end, 15.0.

5. A steam-engine cylinder received steam at 140 lb. per sq. in. gage and exhausted the steam into a vacuum of 25.3 in. of mercury when the barometer was 29.6 in. of mercury. Find the theoretical m.e.p. within the cylinder when the

average point of cut-off was at $\frac{1}{4}$ of the stroke. Find the actual m.e.p. if the diagram factor was 0.85.

6. A steam engine operated during a test with an average m.e.p. of 79.2 lb. per sq. in. when the initial steam pressure was 135 lb. per sq. in. gage and the exhaust pressure was 20 in. of mercury vacuum when the barometer was 29.0 in. of mercury. Find the diagram factor if cut-off occurred at $\frac{3}{8}$ of the stroke.

7. A 12-in. by 12-in. double-acting steam engine with a piston rod $1\frac{3}{4}$ in. in diameter operated at 300 r.p.m. when the initial steam pressure was 185 lb. per sq. in. abs. and the exhaust pressure was 20 lb. per sq. in. abs. Find the theoretical horsepower if the cut-off was at $\frac{5}{8}$ of the stroke.

8. The expansion ratio of a 11-in. by 10-in. double-acting steam engine was 3. Steam at 100 lb. per sq. in. gage was supplied, and the exhaust pressure was 1 lb. per sq. in. gage. Barometer, 29.53 in. of mercury. Engine speed, 200 r.p.m. Find the theoretical i.hp. of the engine if the piston rod area is neglected. Find the actual i.hp. if the diagram factor is 0.6.

9. A 19-in. by 21-in. simple steam engine had a piston rod 4 in. in diameter and operated at a speed of 207 r.p.m. The areas of the indicator diagrams were: head end, 2.47 sq. in.; and crank end, 2.45 sq. in. The scale of the indicator spring was 60 lb. per sq. in., and the average length of the indicator diagrams was 3.67 in. Find the engine constants and the indicated horsepower developed.

10. A 15-in. by 18-in. simple steam engine with a 3-in. piston rod operated at a speed of 225 r.p.m. The i.hp. developed in the head end of the cylinder was 157 and in the crank end 143. Find the m.e.p. in each end of the cylinder. Find the engine constants.

11. A 24-in. by 33-in. steam engine with a piston rod $4\frac{1}{2}$ in. in diameter ran at a speed of 150 r.p.m. The average areas of the indicator diagrams taken with an 80-lb. spring were: head end, 3.12 sq. in.; crank end, 3.27 sq. in. Average length of the diagrams, 3.01 in. Find the total i.hp. Calculate the engine constants.

12. An 8-in. by 12-in. steam engine with a piston rod $1\frac{5}{8}$ in. in diameter ran at a speed of 240 r.p.m. The indicator diagram data were: area of head-end diagram, 2.15 sq. in.; area of crank-end diagram, 2.65 sq. in.; length of indicator diagram, 3.0 in.; spring scale, 80 lb. Find the i.hp. developed.

13. A 10-in. by 10-in. simple steam engine with a piston rod $1\frac{7}{16}$ in. in diameter gave the following test data: speed, 275 r.p.m.; area of head-end diagram, 2.25 sq. in.; area of crank-end diagram, 2.48 sq. in.; length of diagrams, 2.5 in.; spring scale, 100 lb. Find the i.hp. developed.

14. A 7-in. by 7-in. steam engine with a piston rod $1\frac{5}{16}$ in. in diameter gave the following test data: speed, 400 r.p.m.; head-end m.e.p., 31.3 lb. per sq. in.; crank-end m.e.p., 35.8 lb. per sq. in. Find the i.hp. developed.

15. A $3\frac{1}{2}$-in. by $3\frac{1}{2}$-in. steam engine with a piston rod one inch in diameter operated at a speed of 750 r.p.m. Find the total i.hp. when the head-end m.e.p. was 21 and the crank-end m.e.p. was 24.8 lb. per sq. in.

16. Calculate the brake constant for a Prony brake whose arm is $31\frac{1}{2}$ in. long.

17. Calculate the brake constant for a Prony brake whose arm is 63 in. long.

18. How many brake horsepower were developed by a steam engine fitted with a Prony brake $31\frac{1}{2}$ in. long when the gross load on the brake scale was 575 lb.; brake tare, 60 lb.; and the engine speed, 330 r.p.m.?

19. A steam engine running at 150 r.p.m. delivered 200 b.hp. Effective length of the brake arm, $5\frac{1}{4}$ ft. Brake tare, 157 lb. Find the total load on the brake scales.

20. A steam engine developing 509 i.hp. was direct connected to a generator delivering 315 kw. electrical output with a generator efficiency of 92 per cent. Find

(*a*) the mechanical efficiency of the engine, (*b*) the mechanical efficiency of the unit, and (*c*) the friction horsepower of the engine.

21. A steam engine was used to drive, by means of a belt, a generator delivering 250 kw. If the generator efficiency was 90 per cent, the belt efficiency 96 per cent, and the engine mechanical efficiency 87 per cent, what was the i.hp. of the engine? What was the friction horsepower of the engine?

22. A 7-in. by 8-in. steam engine developed 30 b.hp. at full load at a speed of 325 r.p.m. The length of the brake arm was 4 ft 6 in., and the brake tare was 25 lb. Find the scale loads at $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, full, and $1\frac{1}{4}$ loads.

23. Find the brake horsepower and the mechanical efficiency of a 10-in. by 10-in. steam engine with a piston rod $1\frac{1}{16}$ in. in diameter which gave the following test data: speed, 300 r.p.m.; area of head-end diagram, 2.45 sq. in.; area of crank-end diagram, 2.77 sq. in.; length of diagrams, 3.35 in.; spring scale, 80 lb.; brake arm, 5 ft. 3 in. long; gross load on brake scales, 285 lb.; brake tare, 65 lb.

24. Find the brake horsepower and mechanical efficiency of a 10-in. by 12-in. steam engine with a $1\frac{1}{2}$-in. piston rod, which gave the following test data: speed, 195 r.p.m.; area of head-end diagram, 3.9 sq. in.; area of crank-end diagram, 3.72 sq. in.; spring scale, 60 lb.; length of diagram, 3.90 in.; gross brake load, 302 lb.; brake tare, 85 lb.; length of brake arm, 26 in.

25. Find the i.hp. and mechanical efficiency of a 8-in. by 18-in. steam engine with a piston rod $1\frac{7}{8}$ in. in diameter. A test gave the following data: gross brake load, 900 lb.; brake tare, 100 lb.; area of head-end diagram, 3.15 sq. in.; area of crank-end diagram, 2.75 sq. in.; spring scale, 80 lb.; length of diagrams, 3.10 in.; engine speed, 105 r.p.m.; brake arm, 39 in. long.

CHAPTER XII

VALVES, VALVE GEARS, AND GOVERNORS

SLIDE-VALVE ENGINES

177. Functions of Valves.—The purpose of steam-engine valves and the gear which actuates them is to control the distribution of steam within the engine cylinder. The distribution process includes the control of the events of admission, cut-off, expansion, release, exhaust, and compression of the steam during each working cycle. Both the power developed by the engine, and the economy of operation, are affected by the valve and the valve-operating mechanism. Certain valves, together with their gear, have inherent limitations which materially reduce the thermal efficiency of the engine thus equipped. Other engines equipped with a better valve mechanism, and operating under identical conditions, show considerable savings in the amount of heat energy that must be supplied to produce a given amount of mechanical power.

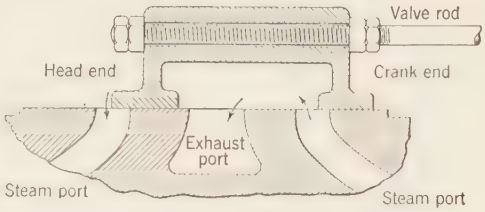


FIG. 149.—Simple D slide-valve.

178. The D Slide-valve.—One of the simplest and most easily constructed of all steam engine valves is the D slide-valve. The designation D is applied because of the sectional shape of the valve as shown by Figs. 138 and 149. The valves as illustrated are allowing admission of steam to the head end and exhaust of steam from the crank end of the cylinder.

Such valves are used in engines of simple design and where the first cost of the engine is of prime importance. Valves of this type, when used singly, do not permit cut-off within the engine cylinder at less than about five-eighths of the stroke without serious interference with the other events of the cycle. If attempts are made to secure earlier cut-off with a single valve controlling all the events, the power output is reduced and smooth operation of the engine is prevented. The comparatively late cut-off thus secured limits the number of times that steam may be expanded within a single cylinder and reduces the heat economy of the engine.

Where exhaust steam can be used for feedwater heating, process work, or the heating of buildings, engines equipped with slide-valves and operating with high water rates may be used with economical results. The engine producing mechanical power acts as a rotary pressure-reducing valve, and the heat rejected to the exhaust is utilized for some useful purpose. When disposal of exhaust steam is made in this manner, mechanical power may be secured at reduced costs because of the market value of the exhaust steam. The assumption must not be made that the mechanical power secured is produced without cost even though live steam from the boilers would be required for the heating demands if exhaust steam were not available. The power production must be charged with certain initial, maintenance, interest, depreciation, and operating expenses.

Definite terms of nomenclature are applied to all valves. These terms are most readily explained by the use of sections of a D valve, but the terms of the following articles are applicable to other valves.

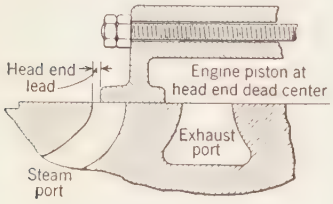


Fig. 150.—Valve lead.

179. Lead.—Admission of steam to the cylinder usually occurs just prior to the dead-center position of the piston. With admission ahead of dead-center position the valve uncovers the port near the end of the piston stroke. Lead is the amount by which the valve uncovers the steam port when the engine is at the dead-center position, Fig. 150. The measurement is made in terms of inches. Lead is given to an engine to aid in cushioning the piston as it comes to rest, and to reduce wire-drawing losses as the piston starts on its outward stroke.

Leads may be made equal at each end of the cylinder by adjustment of the valve gear. When the leads are equal at both ends of the cylinder the cut-offs are unequal, with the ordinary slide-valve gear.

180. Lap.—Two designations of lap are made, namely, *steam lap* and *exhaust lap*. Measurements of steam or exhaust laps cannot be correctly made until the valve is in its mid-position of travel. A valve is in mid-position when it is half way between the extremities of its travel. Mid-position of the valve changes with every change of the valve setting, and the valve is not always in mid-position when its center line coincides with the center line of the valve seat.

Steam lap is the amount by which the admission edge of the valve overlaps the admission edge of the port when the valve is in mid-position, Fig. 151. Exhaust lap is the amount by which the release edge of the valve overlaps the release edge of the port when the valve is in mid-position, Fig. 151. The steam and the exhaust laps are not usually

equal for the two cylinder ends, especially if the cut-offs are equal. The laps are necessarily unequal owing to the angularity of the connecting rod and the eccentric rod. Greater travel of the piston and the valve occurs in the first quarter of the revolution, in which the piston moves from the head end. The angularity of the eccentric rod is of small effect, but that of the connecting rod is appreciable. This angularity of the connecting rod requires that a greater steam lap be given to the head end of the valve than is given to the crank end.

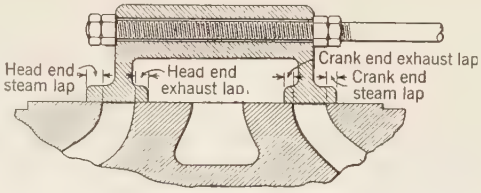


Fig. 151.—Steam and exhaust laps.

Steam laps are necessary in order to give cut-offs earlier than the end of the stroke. A valve without lead or laps is shown by Fig. 152, and cut-off occurs at the end of the stroke. Valves of this type are frequently used in direct-acting pumps, Fig. 238.

Exhaust laps aid in giving compression to the steam near the end of the return stroke, although some automatic high-speed engine valves have no exhaust lap. In some cases the valve does not completely cover the port on the exhaust side when in mid-position. This latter condition is called *exhaust clearance.*

181. Angle of Advance.—The effect of an eccentric, Fig. 153, is that of a crank which moves the valve back and forth as the eccentric is rotated. This effect is obtained by making the eccentric in the form of an enlarged crank pin whose center does not coincide with that of the shaft upon which it is mounted. The eccentric turns about the center of the crankshaft and not about its own geometrical center. The distance between the center of the crankshaft and the center of the eccentric is called the *eccentricity*, Fig. 153, and is equal to the crank arm effect of the eccentric. The motion imparted to the valve slide and valve is equal to twice the eccentricity.

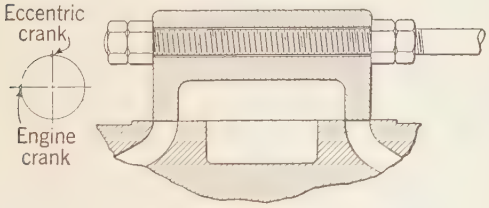


Fig. 152.—Slide-valve without laps.

If the valve had no laps and no lead, the effective crank arm of the eccentric would be set ahead of the engine crank by an angle of 90 deg. However, the valve must be moved to give the necessary lead at dead center, which necessitates that the eccentric be ahead of the crank by an angle equal to (90 deg. + α), Fig. 154. The angle a is the lap angle, as the eccentric must be advanced by this amount to over-

come the steam lap. As the valve also has steam lead, in order to give the proper timing of events in the cycle, the eccentric is turned ahead of the crank by a further angle of b, which is the lead angle. The angle α, which is the sum of two angles a and b, is the angle of advance. The

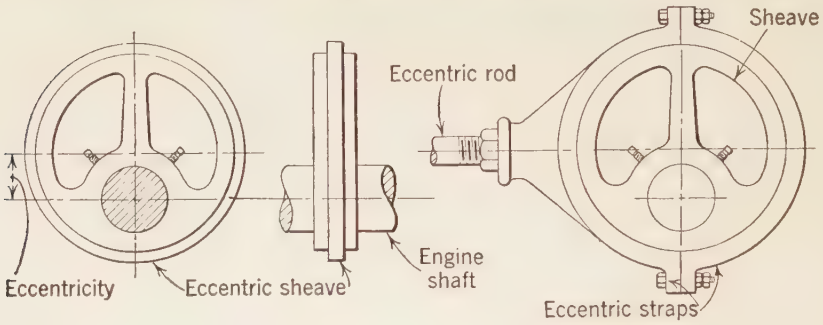


FIG. 153.—Eccentric sheave and straps.

angle of advance may also be stated as the total angle between the eccentric crank and the engine crank minus 90 deg.

182. Valve Diagrams.—The action of existing slide-valves may be studied and the design of new valves facilitated by the use of graphical constructions known as valve diagrams. The function of any such construction is to show the displacement of the valve from its mid-position at any time during a revolution of the engine crankshaft. From certain known data, unknown data relative to the valve and unknown positions of events of the cycle may be determined. Common valve diagrams are the Zeuner and the Bilgram, both of which have their uses and limitations. The Bilgram diagram is simpler for some designs, but the Zeuner tends to give a better pictorial representation of the valve action.

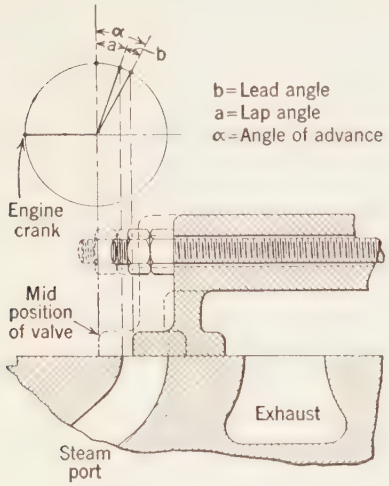


FIG. 154.—Angle of advance.

183. The Zeuner Diagram.—The Zeuner diagram is constructed in the following manner, beginning with Fig. 155. Lay off a pair of axes XX' and YY' at right angles to each other. With O as a center describe a circle the diameter, XX', of which represents to some scale

the stroke of the engine and to another scale the valve travel. The engine crank pin and the eccentric are assumed to be rotating in a clockwise direction, or the engine is running over when the horizontal cylinder is to the left of X the head-end dead-center position of the crank pin.

Take OF as the position of the engine crank at any position beyond the head-end dead center in the direction of rotation. The corresponding position of the eccentric crank will be $(90 + \alpha)$ deg. ahead of the engine crank or along the line OE. Therefore, the displacement of the valve from its mid-position is represented by yE as the mid-position of the valve will occur when the eccentric is at OY. Then from the point O locate on OE the point f so that Of is equal to yE. When the eccentric is at OE', the valve displacement is $y'E'$, and the point f' is located by making Of' along OE' equal to $y'E'$. Likewise, the valve displacements for eccentric positions OY and OX' are equal to zero and OX', respectively. Any number of eccentric positions may be taken and the valve displacements laid out along the line of the eccentric crank. Through the points so located a circle is drawn with the line OX' as a diameter. This circle is known as the right valve circle as it is used to find the valve displacements to the right of mid-position for various eccentric positions. The left valve circle is developed upon the line OX in a similar manner and is used to indicate the valve displacements to the left of mid-position.

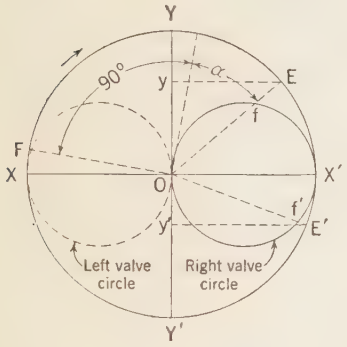


Fig. 155.—Zeuner valve-circle construction.

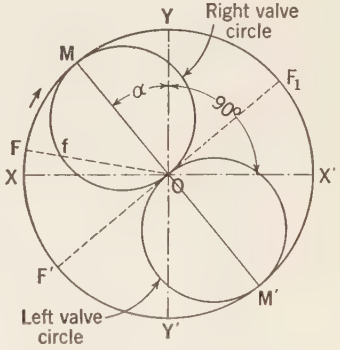


Fig. 156.—Valve-circle rotated $(90+\alpha)$ degrees.

If the line OE is revolved, about O as a center, $(90 + \alpha)$ deg. in a counter-clockwise direction, until it coincides with the line OF, the conditions of Fig. 156 will result and angle YOM is equal to α. Again Of represents the valve displacement for the crank position OF. The intersections of the engine crank lines with the right and left valve circles

may be used to find the valve displacements, from mid position, to the right and to the left for various crank pin positions. When the engine crank is at either OF_1 or OF' the valve displacement is zero and the valve is in mid-position; the maximum displacements each side of mid-position occur when the crank positions are OM and OM', respectively, and at these points the valve changes its direction of motion from right to left and from left to right.

The net opening of the steam port is the valve displacement minus the steam lap. Consequently, if the arc of a circle with a radius equal to the head-end steam-lap s be described about O as a center, the steam-port opening for crank position OF is fb, Fig. 157. The intersection d of the line OA and the right valve circle, Fig. 157, represents the condition where the steam port is about to be uncovered, and the engine crank position at OA is that necessary for the admission of steam to the head end of the cylinder. When the crank pin is at X the steam-port opening is equal to the head-end lead an which is also equal to the radius of a circle whose center is at X and which is tangent to the line AC at l. As the engine crank passes through various positions in its rotation the head-end steam-port opening increases to a maximum at OM and decreases to zero at the position OC where head-end cut-off occurs.

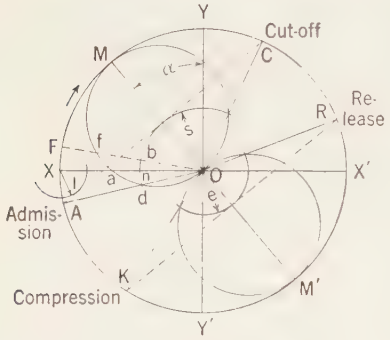


Fig. 157.—Zeuner valve-diagram for head-end.

Using O as a center, an arc should now be described with a radius equal to the exhaust lap e, cutting the left valve circle. This construction will be utilized to determine the exhaust events of the head end of the cylinder. When the engine crank reaches the position OR, the valve is to the left of its mid-position by an amount equal to the head-end exhaust-lap, and the event of release occurs. Further movement of the engine crank causes the exhaust port to open to a maximum at OM' and to become zero at position OK, where the exhaust port is just closed and compression of the steam begins. In the geometrical construction the line MM' bisects the angles AOC and ROK. Therefore the lines AC and RK are at right angles to the line MM', are parallel to each other, and are tangent to the steam and exhaust-lap arcs, respectively. These facts are of use in solving some valve-diagram problems.

The representation thus far is for the cycle of events in the head end

of the engine cylinder. A study of the crank-end events is made in a similar manner and is shown by Fig. 158.

184. Uses of the Zeuner Diagram.—The customary practice is to combine both the head-end and crank-end diagrams into a single figure, which has been done in Fig. 159 by superimposing Fig. 158 upon Fig. 157.

Owing to the angularity of the connecting rod, the piston travel is not at a uniform rate throughout a revolution of the crankshaft. The same is true in the case of the valve, but as the ratio of the eccentricity of the eccentric to the length of the eccentric rod is small the effect produced by angularity is so slight that it may be neglected. However,

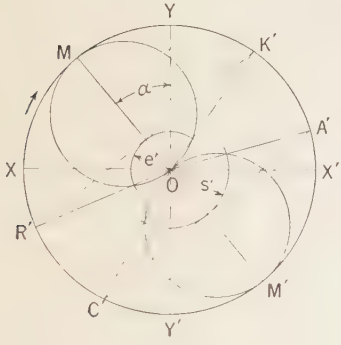


Fig. 158.—Zeuner diagram for crank-end.

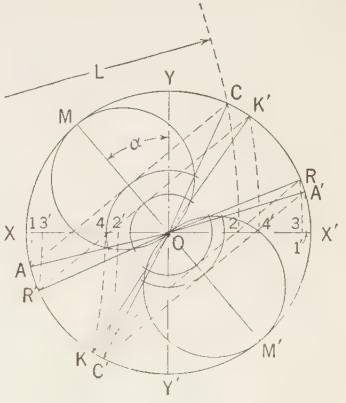


Fig. 159.—Zeuner diagrams for both head- and crank-ends, corrected for angularity of the connecting rod.

in the case of the connecting rod its effect cannot be ignored. Therefore, to find the piston positions for the various events of a cycle the angularity of the connecting rod must be considered.

The true positions of the engine piston are located by striking arcs through the points A, C, R, K, etc., to cut the line XX' at 1, 2, 3, 4, etc. The radius of each of these arcs is equal to the length of the connecting rod L, to the scale of the drawing, and their centers are located upon the line XX' extended to the left. The portion of the forward stroke completed for head-end cut-off, Fig. 159, is equal to the length of line $X2$ divided by the length of the line XX'. Expressed as a percentage this becomes $100\ (X2) \div XX'$. Likewise the other piston movements for the different events may be expressed as percentages of the engine stroke provided the measurements are made from the proper end of the line XX'.

185. Slide-valve Setting.—Either of two methods may be employed in adjusting a valve gear to give the best distribution of steam within an

engine cylinder. The adjustments may be made in conjunction with the use of an engine indicator or they may be made by measurement. Valve setting by use of an indicator is simpler, as all adjustments are made outside of the steam chest, which is closed. The adjustments consist merely in lengthening or shortening the valve rod and changing the eccentric position. Indicator diagrams disclose the effect of the changes made, and when the adjustments are made to give the best diagram possible, the setting is completed.

186. Effects of Slide-valve Gear Changes.—When the engine has a simple valve gear as shown by Fig. 137, the eccentric crank is set ahead of the engine crank, in the running direction, by an angle of $(90 + \alpha)$ deg., where α is the angle of advance. The effect of advancing the eccentric or making α greater is to make all the events of the cycle come earlier for both ends of the cylinder. Retarding the eccentric has the opposite effect or makes all events later in the cycle. The adjustments made upon the valve consist merely in lengthening or shortening the valve rod. This change may be made by moving the valve on the valve rod or by changing the effective length of the valve rod at the valve-rod slide and guide. Any adjustment of the valve or valve rod which moves the valve toward the crank end of the cylinder shortens the valve rod. Any change of the valve position toward the head end of the cylinder has the effect of lengthening the valve rod. Irrespective of how the changes of valve position are accomplished the alterations thus produced will hereafter be designated as having been accomplished by lengthening or shortening the valve rod. An inspection of the items of Table XVIII reveals that changing the length of the valve rod produces opposite effects at the two ends of the cylinder.

Fig. 160.—Trammel rod.

187. Location of Engine Dead Centers.—Engine dead-center positions can be rapidly located by the use of a trammel rod, Fig. 160. The engine piston is placed slightly ahead of dead center and corresponding marks are made on the crosshead and one of the guides. Using some convenient point on the engine frame or floor as a reference point, a mark is scribed by the trammel on the engine flywheel. The engine piston is then moved in its running direction beyond the dead-center position until the marks on the crosshead and guide coincide. Another arc is now scribed on the flywheel by the trammel from the same reference point. The linear distance on the flywheel between the scribed arcs is bisected by the aid of dividers, and the mid-position thus secured is brought under the free end of the trammel whose opposite end rests on the original reference mark. In locating dead centers the engine

should always be turned in its running direction to avoid any lost motion in the moving parts.

TABLE XVIII

EFFECTS OF CHANGES IN LENGTH OF VALVE ROD

Change	Events Made		Lead		Laps	
	Later	Earlier	Greater	Less	Greater	Less
Valve Rod Lengthened	H. E., Adm. H. E., Comp. C. E., C. O. C. E., Release	H. E., C. O. H. E., Release C. E., Adm. C. E., Comp.	C. E.	H. E.	H. E. Steam C. E. Exhaust	C. E. Steam H. E. Exhaust
Valve Rod Shortened	C. E., Adm. C. E., Comp. H. E., C. O. H. E., Release	C. E., C. O. C. E., Release H. E., Adm. H. E., Comp.	H. E.	C. E.	C. E. Steam H. E. Exhaust	H. E. Steam C. E. Exhaust

188. Slide-valve Setting by Measurement.—Valve setting by measurement is common practice when an engine indicator together with a reducing motion are not available. The first step in the procedure is to locate accurately both the head-end and crank-end dead centers. The linear distance between the two dead-center positions is the actual stroke of the engine. The valve setting by measurement may be made for equal cut-offs or equal leads, but not both at the same time.

Procedure for equal cut-offs.—Lay off from both the head and the crank ends on the crosshead guide linear distances representing the percentages of the stroke desired at cut-off. These measurements should be laid off from the dead center in the direction of the piston motion. Put the engine on dead center and shift the eccentric so that the angle of advance is about 45 deg. Turn the engine in a running direction until the desired portion of the stroke at cut-off is completed. Remove the steam-chest cover and lengthen or shorten the valve rod until the cutting-off edge of the valve just coincides with the cutting-off edge of the port at the cylinder end under consideration. Turn the engine in the running direction until the piston is at the desired cut-off position at the opposite end of the cylinder. Measure the amount by which the valve covers or fails to cover the cut-off edge of the port for this end of the cylinder. Lengthen or shorten the valve rod, as the case may be, to correct one-half of the error, and shift the eccentric on the shaft to eliminate the other half of the error. Put the engine piston at the cut-off position at the opposite end of the cylinder, and remove one-half of any error by adjusting the length of the valve rod and the other half by moving the eccentric on the shaft.

Set the engine first on one dead center and then the other, and measure the lead in each case. Measure the position of the valve center on the seat when the valve is in an extreme position of movement. From prior measurements of the valve, the valve seat, and the port widths calculate the head- and crank-end steam and exhaust laps as well as the maximum port openings for each end of the cylinder. See that all nuts are locked tightly, and replace the steam-chest cover. If an indicator is available apply a normal load to the engine, and take a diagram for each end of the cylinder. Make such minor adjustments as are necessary.

Setting for equal leads.—Place the engine on dead center with the angle of advance approximately 45 deg. Move the valve to give the desired lead. Place the engine on the opposite dead center and secure the desired lead by lengthening or shortening the valve rod to remove one-half the error and shifting the eccentric to correct the other half of the error. Check the lead again at the opposite end of the cylinder and remove one-half of any error by moving the valve and the eccentric, respectively. Find the percentages of the stroke at which the cut-offs occur as well as the steam and exhaust laps and maximum port openings.

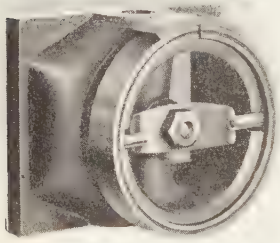


Fig. 161. Partially balanced slide-valve.

189. Balanced Valves.—The slide-valve shown by Fig. 149 is unbalanced in that the pressure within the steam chest forces the valve against the seat upon which it moves. Under such conditions a considerable amount of power may be required to move the valve (especially large valves), and difficulty is encountered in the lubrication of the faces of the valve and the valve seat.

The steam pressure on the back of the valve may be partially relieved in several ways. One method of pressure relief is as shown by Fig. 161 where the valve is made to slide between the steam-chest cover and the seat. A packing ring is fitted to the back of the valve and held against the steam-chest cover by the action of a spring. The steam pressure is thus relieved from the greater portion of the back of the valve and the frictional load on the valve gear reduced. The valve thus fitted is only partially balanced, as sufficient pressure is maintained on the back of the valve to hold it firmly against the valve seat.

190. Piston Valves. The piston type of valve, Fig. 162, is in principle a D slide-valve. Valves of the piston type are balanced as the steam pressure is equally exerted upon all sides. Piston valves move back and forth in a cylindrical valve chamber which confines them.

This confinement prevents their lifting from the seat when compression of an excessive amount of water in the clearance space of the engine occurs. Damage to the engine may result under such conditions.

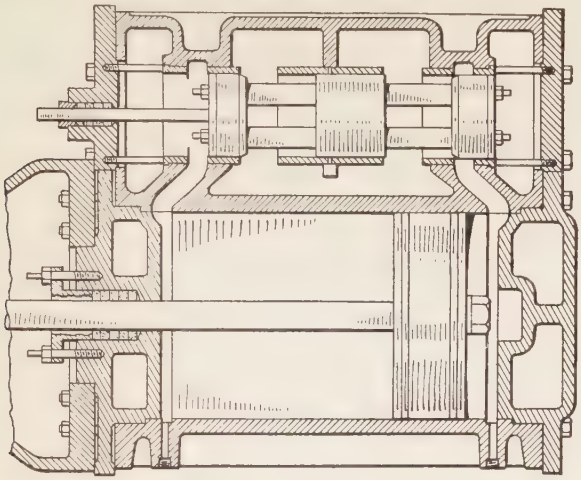


FIG. 162.—Piston valve.

191. Double-ported Valves.—Single steam ports at the cylinder ends may require excessive valve travel in order that the port be fully opened during the admission period. Sluggish valve action may cause considerable wiredrawing during the admission period. In order to eliminate such action, double steam ports to the cylinder ends or ports through the valve or both may be provided. A ported valve is shown by Fig. 163, which illustrates a Sweet valve. The valve illustrated allows by a small movement a greater port opening. The Sweet valve is also partially balanced as it moves between a pressure plate and the valve seat.

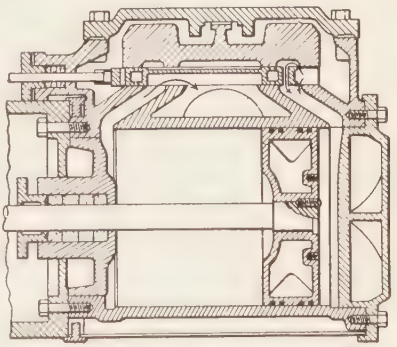


FIG. 163.—Sweet valve, Ames steam engine.

192. Speed Regulation.—Ordinarily steam engines are provided with a governing mechanism which functions to maintain a nearly constant speed of the engine. The purpose of the governor is to limit or increase the power input of the engine as the load decreases or increases. The regulation must be very close in certain classes of work, as, for instance, engine-driven electrical generators where the voltage varies

directly as the engine speed. Governors must also serve to limit excessive and dangerous engine speeds.

A governor must be stable in order to give satisfactory regulation. The term stable signifies that, for any predetermined speed, the governor will assume a definite position. Any variation from the predetermined speed will cause the governor to move to bring the speed back to normal. The governor movements must be regular and certain without excessive speed fluctuations. Unstable governors move rapidly from one speed to another without being able to control the engine at any fixed rotative speed. When the governor is over-sensitive and the engine speed varies rapidly from low to high and then from high to low, the engine is said to " hunt."

Engines are built to deliver a normal or full load at some designated rotative speed, and the governor should be adjusted to give the normal speed. The *coefficient of regulation* is the percentage of variation of the rotative speed from the normal or full-load speed. The coefficient of regulation expressed as a percentage is

$$K_R = \frac{(N' - N)100}{N}, \tag{119}$$

in which K_R = coefficient of speed regulation, per cent.
N = normal full-load speed, r.p.m.
N' = no-load speed, r.p.m.

For electric lighting service K_R should not exceed $1\frac{1}{2}$ per cent for slow changes, and 2 per cent for sudden load changes.

Speed regulation of engines is accomplished by two methods of governing: *throttling* and *cut-off*. The former method effects the regulation by varying the initial steam pressure; the latter is often but not always accomplished by some form of a shaft or flywheel governor, which varies the volume of steam admitted to the cylinder. Releasing gear Corliss engines have variable cut-off without shaft governors.

193. Throttling Governor.—This type of governor is employed with reciprocating engines having a fixed point of cut-off, small steam turbines, and air compressors.

The construction of a typical throttling governor is shown by Fig. 164. A balanced valve A is carried upon the lower end of the valve spindle B, which passes upward through a stuffing box at the top of the governor body. The upper end of the spindle B is rigidly attached to a speed-changing device C. Above the speed-changing device is another spindle, D, which passes upward through a revolving sleeve E held within a vertical bearing. The sleeve is driven through a pair of miter gears F, F, from a short shaft supported by a bracket. The rotative

motion of the shaft is derived from the main shaft of the engine by means of a belt and pulleys. The upper stem D is centered in its contact with the speed changer and does not revolve.

Governor balls or weights, G, G, G, each made in two pieces, are fastened to three flat steel springs, I, I, I, by means of a central stud. The springs are rigidly attached to the revolving sleeves P and R, making a unit that actuates the valve. The cap K, screwed into the recess of the revolving sleeve R, bears on the upper valve stem button S and is employed for depressing the governor valve. When the engine is not run-

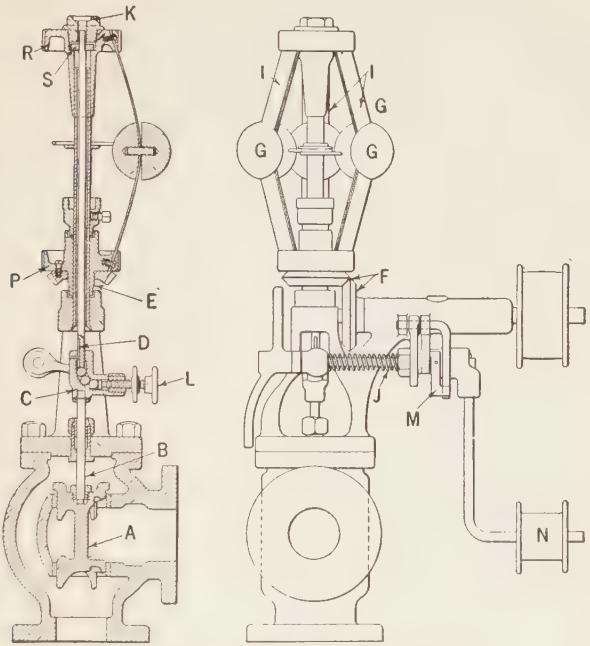


Fig. 164.—Pickering throttling governor.

ning, the balanced valve A is held open by the action of a coil spring J, which may be adjusted for various spring tensions, according to the individual conditions of the engine.

The action of the governor is procured as the revolving flyballs are thrown outward by centrifugal forces. The movement thus secured causes the cap K to push the stem D downward, and thus partially close the valve A against the action of the coil spring J. This action causes a restriction in the amount and the pressure of the steam admitted to the steam chest of the engine or turbine. A further increase of engine speed causes the governor balls to rotate more rapidly and to be moved farther

outward, thus further closing the valve A against the action of the spring. A decrease of engine speed has the opposite effect.

Engine speed changes may be effected by means of an adjusting screw L, which allows hardened-steel balls to be inserted under or removed

FIG. 165.—Action of a throttling governor.

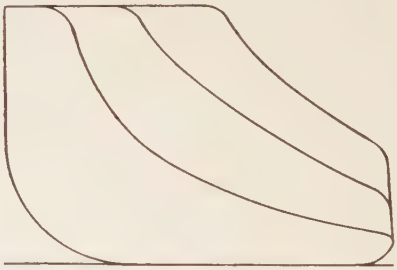


FIG. 166.—Action of a cut-off governor.

from directly underneath spindle D. The effect of the balls is to lengthen or shorten the stem D and thus cause the valve A to be opened or closed more or less for a given position of the governor weights or flyballs. The governor is fitted with a safety device M, so that if the governor belt should break the idler pulley or belt tightener N will drop and by means of a ratchet and pawl arrangement will close the valve A and thus stop the engine or turbine. The effect of a throttling governor with varying engine loads is shown by Fig. 165.

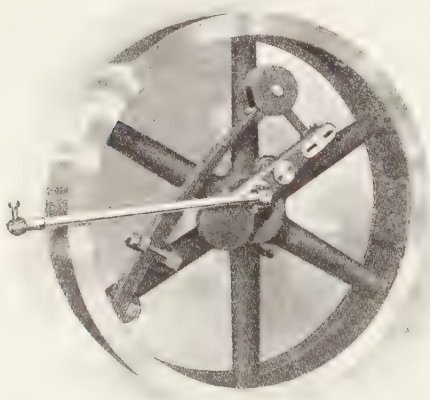


FIG. 167.—Robb-Armstrong-Sweet centrifugal-inertia governor, Ames steam engine.

194. Flywheel or Shaft Governors.—Engines whose speed is controlled by the duration of the steam admission period usually have cut-off for full or normal loads at $\frac{1}{4}$ to $\frac{1}{3}$ the stroke. When speed increases occur through decreased load, the governor functions to make the cut-off earlier; and when heavy loads decrease the speed, cut-off comes much later in the stroke, Fig. 166. The control of the point of cut-off is often accomplished by shaft or flywheel governors of the following types: (1) centrifugal, (2) inertia, and (3) combination of

centrifugal and inertia. As a matter of fact, it is very difficult to build a flywheel governor that does not embody some of the characteristic forces of both the centrifugal and inertia types.

Flywheel governors vary the point of cut-off by changing: (1) the angle of advance of the eccentric and the amount of valve travel, (2) the eccentricity of the eccentric and the amount of valve travel; and (3) both the angle of advance and the valve travel.

195. Centrifugal Governor.—Figure 167 illustrates a Robb-Armstrong-Sweet centrifugal governor as built into the flywheel of an Ames single-valve, center-crank steam engine. The mechanism consists of a leaf spring rigidly attached at one end to a spoke of the flywheel. The free end of the spring carries a weight which is connected by a link to a lever weighted at one end. The weighted lever is pivoted near the flywheel hub at about the middle of the lever. Between the lever weight and the lever pivot point an eccentric rod is attached to the lever. This eccentric rod transmits motion to a valve rocker and thus gives motion to the engine valve.

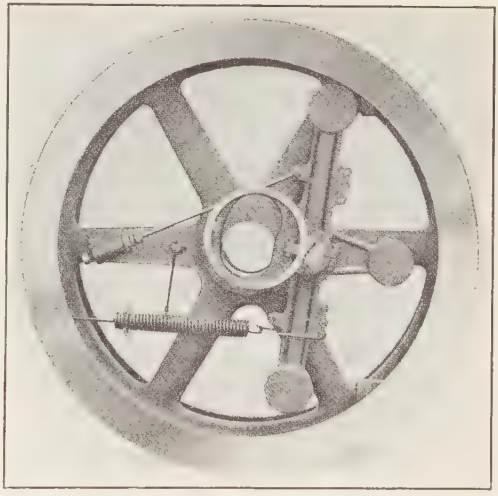


FIG. 168.—Inertia governor of an Elliott steam engine.

In the position shown, the governor weights rest near the center of the wheel, due to the action of the spring. As the engine acquires speed, centrifugal forces move the weights outward against the spring resistance. When the weight movement occurs the weighted lever is moved about its pivot and the point of attachment of the eccentric rod is shifted with respect to the center of the engine crankshaft. The action of the governor is to change both the angle of advance and the amount of the valve travel. The operating speed of the engine is changed by moving the point of attachment of the link at the weighted lever.

196. Inertia Governor.—A simple form of inertia governor is shown by Fig. 168, which illustrates the flywheel and governor of an Elliott engine. The governor embodies a weighted lever pivoted at its center on one of the flywheel spokes. When the engine is at rest, the action of a coil spring brings one of the weights against a stop as shown at the top

of the wheel. A swinging eccentric is rigidly attached to the weight lever and rotates with the lever about the pivot point. The engine shaft passes through a slot of the eccentric. so that the eccentric is entirely free from the engine shaft.

When the engine is at rest the eccentric is at a position necessary to give the maximum cut-off. As the engine speeds up, the rim of the flywheel tends to run ahead of the governor weights which lag behind

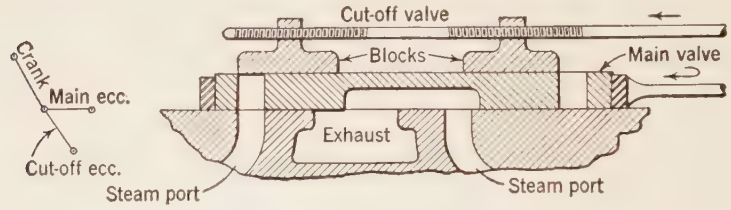


Fig. 169.—Riding cut-off valve at admission.

on account of inertia forces. The result is that the eccentric position is shifted with respect to the engine crank and the point of cut-off is made earlier. When the flywheel tends to run slower the governor weights crowd ahead, owing to inertia forces, and the eccentric is shifted to give later cut-off. Speed may be changed by changing the tension of the coil spring. The oil cylinder and rod attached to the upper end of the weight lever are for the purpose of dampening the action of the governor and thus preventing over-sensitiveness.

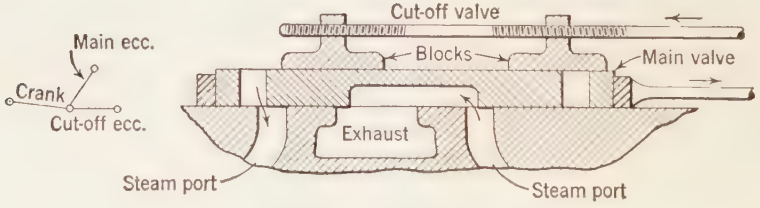


Fig. 170.—Riding cut-off valve at cut-off.

197. Riding Cut-off Valves.—Auxiliary valves are sometimes applied to D slide-valves to control the point of cut-off independent of the other events of the stroke. In this manner one of the serious limitations of a D slide-valve may be eliminated. Illustrations of the action of a slide-valve equipped with a riding cut-off are shown for the events of admission and cut-off, by Figs. 169 and 170. The main valve is ported and normally is set to give cut-off at about $\frac{3}{4}$ the engine stroke. The main valve is driven by an eccentric rigidly attached to the engine shaft. The rider or cut-off valve is usually driven and controlled by the mechanism

of a shaft governor. The sole function of the rider valve is to give earlier cut-off than is possible with the main valve, without interference with the events of admission, release, and compression. Concentric piston valves may also be used to embody the riding cut-off principle.

198. Automatic High-speed Engine.—A typical automatic high-speed engine is illustrated by Fig. 171. This engine is of the center-crank type as the engine crank is midway between the main bearings of the engine frame. The engine is equipped with a Sweet balanced

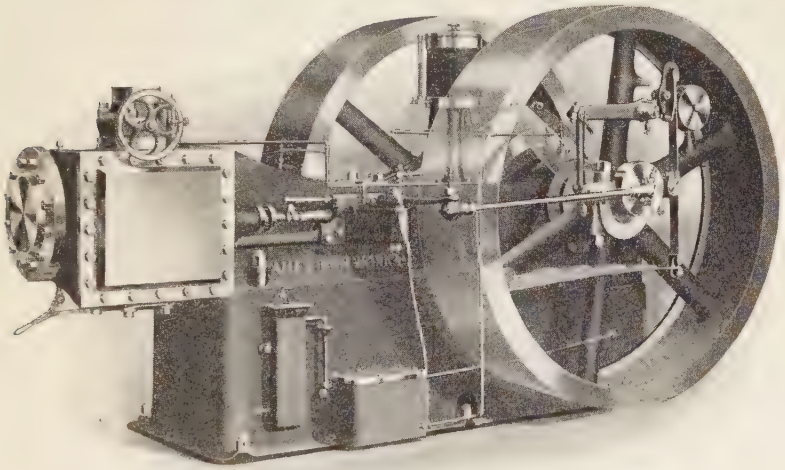


Fig. 171.—Ames automatic high-speed slide-valve engine.

slide-valve and a centrifugal governor. The valve motion is transmitted from the governor parts by the eccentric rod to a horizontal rocker, which moves the valve slide by a slotted crosshead. Engines of this type may be obtained in sizes ranging from 30 to 400 hp.

PROBLEMS

1. A 10-in. by 12-in. steam engine with a connecting rod 36 in. long has a valve travel of 3 in. and an angle of advance of 40 deg. The lap data are: head-end steam, $\frac{3}{4}$ in., and exhaust, $\frac{5}{16}$ in.; crank-end steam, $\frac{9}{16}$ in., and exhaust, $\frac{1}{2}$ in. Find the percentages of the stroke at which the events of the cycle occur for both head and crank ends. Make the figure 3 in. in diameter.

2. A 7-in. by 10-in. steam engine has a connecting rod 30 in. long and a valve travel of 2 in. The valve is to be set for equal cut-offs at $\frac{5}{8}$ of the stroke, and the compression is to start at 85 per cent of the return stroke in each cylinder end. Head-end lead is $\frac{1}{16}$ in. Find, for both cylinder ends, the piston positions at admission and at release, the steam and exhaust laps, the maximum port openings, the angle of advance, and the crank-end lead. Draw the eccentric circle 4 in. in diameter.

3. An engine has equal leads of $\frac{3}{16}$ in. for each cylinder end. The head-end exhaust lap is $\frac{3}{8}$ in., giving release at 95 per cent of the stroke, and the crank-end exhaust lap is $\frac{1}{4}$ in. The engine has a stroke of 18 in., a connecting rod 48 in. long, and a valve travel of 4 in. Find the angle of advance, the steam laps, and the percentages of the stroke at which admission, cut-off, and compression occur in each end of the cylinder.

4. An engine is to have a head-end steam-port opening of $\frac{7}{8}$ in., equal leads of $\frac{1}{16}$ in., release at 95 per cent of the stroke in each cylinder end, and head-end steam lap of $\frac{5}{8}$ in. The length of the connecting rod is to be 3 times the engine stroke. Find the angle of advance, the eccentricity of the eccentric, the steam and exhaust laps, and the percentage of the stroke completed by the piston for the other events of the cycle.

CHAPTER XIII

STEAM-ENGINE ECONOMY

199. Steam Consumption.—Dependent upon the load carried, a steam engine requires a certain total amount of steam per hour which includes not only that used inside of the cylinders but also that used in cylinder jackets and reheater coils. The total weight of steam necessary per hour may be shown graphically by plotting the total amounts used per hour as ordinates against the corresponding engine loads, either as i.hp., b.hp., or kw., as abscissas. The shape of the resulting curve, which is known as the *Willans line*, is dependent upon the method by by which the engine is governed. For throttling-governed engines the curve is nearly a straight line until the engine is overloaded. Engines controlled by automatic cut-off governors usually give curves which are bowed downward with respect to the axis of the abscissa. Such curves when partially completed are of some use in predicting the total steam consumption for loads not already measured.

200. Specific Steam Consumption or Water Rate.—The weight of dry steam consumed by a steam engine per horsepower per hour is the *water rate* of the engine. The expression is made on the basis of either brake or indicated horsepower. The water rate of an engine is dependent upon a number of factors: (1) initial steam pressure; (2) initial steam temperature; (3) exhaust pressure; (4) number of expansions of the steam; (5) number of cylinders used for expansion; (6) method of governing; (7) valves and valve gear; (8) percentage of rating developed; (9) type of engine, counterflow or unaflow; (10) mechanical design and construction; (11) cylinder insulation; and (12) steam jackets and reheaters.

Comparative performances of steam engines cannot be judged by the water rates alone because of the number of factors affecting the engine operation. The relative merits of thermal performance of various engines may be obtained more accurately by comparisons of the amounts of heat supplied to the engines per horsepower-hour. Because of the great diversity of operating conditions the water rates of only a few representative engines are given by Fig. 172.

201. Thermal Efficiency.—Thermal efficiency is the ratio of the output (heat equivalent) to the quantity of heat available to the engine to

produce the power. The ratio or thermal efficiency for saturated steam when computed on the basis of one horsepower is

$$e_t = \frac{2545}{W_s(h_{f,1} + x_1 h_{fg,1} - h_{f,2})}, \tag{120}$$

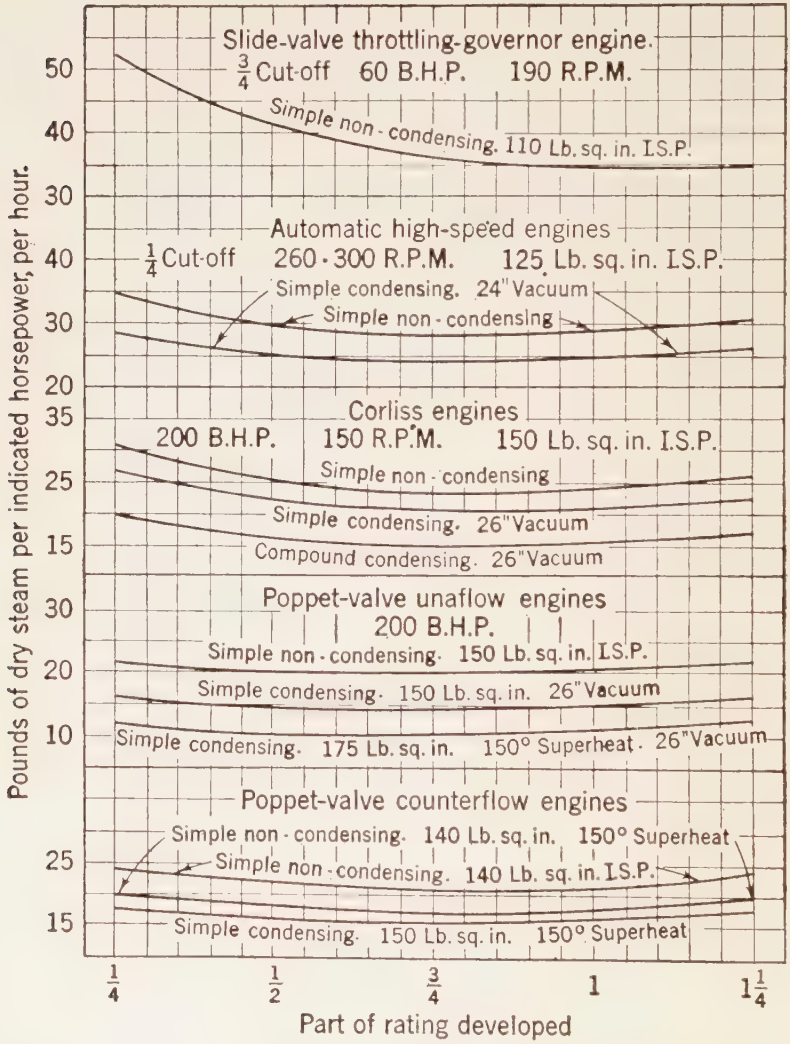


FIG. 172.—Steam consumption of various engines.

where e_t = actual thermal efficiency.

2545 = heat equivalent of one horsepower-hour, B.t.u.

W_s = actual pounds of steam supplied per hp-hr.

$h_{f,1}$ = heat of liquid, initial pressure abs., B.t.u. per lb.
x_1 = quality of the steam supplied.
$h_{fg,1}$ = heat of evaporation, initial pressure abs., B.t.u. per lb.
$h_{f,2}$ = heat of liquid, exhaust pressure abs., B.t.u. per lb.

When superheated steam is supplied to the engine the expression becomes

$$e_t = \frac{2545}{W_s(h - h_{f,2})}. \qquad (121)$$

h = total heat of superheated steam, initial pressure abs., B.t.u. per lb.

All the heat of the steam supplied to the engine above the heat of the liquid corresponding to the temperature at exhaust pressure is considered as available to the engine.

Inspection of equations 120 and 121 reveals that, in either case, when the initial conditions are unchanged, anything which reduces the quantity W_s increases the thermal efficiency. The thermal efficiency varies inversely as the steam consumption per horsepower-hour.

Example.—A steam engine uses 40 lb. of wet steam per b.hp-hr. Initial pressure of the steam is 150 lb. per sq. in. abs., exhaust pressure 15 lb. per sq. in. abs., steam quality 98 per cent. Find the thermal efficiency.

Solution.—

$$e_t = \frac{2545}{40\,[330.4 + (0.98 \times 863.1) - 181]} = 0.064 \text{ or } 6.4 \text{ per cent.}$$

202. Engine Efficiency.—The ideal or Rankine cycle has been set up as the best possible performance for any conditions. The theoretical efficiency for this cycle was discussed in Art. 168. The actual engine performance falls short of the ideal because the conditions of the ideal cycle cannot be fulfilled. The ratio of the actual thermal efficiency to that of the Rankine cycle for the same pressure and temperature conditions is the *engine efficiency*, e_e.

$$e_e = \frac{e_t}{e_R} = \frac{2545}{W_s(h_1 - h_{f,2})} \times \left(\frac{h_1 - h_{f,2}}{h_1 - h_2}\right) = \frac{2545}{W_s(h_1 - h_2)}. \qquad (122)$$

h_2 is the total heat per pound of steam after it has been expanded adiabatically from the initial to the final pressure. This heat may be obtained from the Mollier diagram or it may be calculated as $h_2 = h_{f,2} + x_2 h_{fg,2}$. The final quality of the steam is computed by the method of Art. 45.

The engine efficiency varies from 0.35 for poorly designed and operated engines to 0.70 or 0.80 for high-class, high-efficiency engines.

203. Thermal Losses.—Certain losses occurring in steam engines affect the steam consumption and the thermal efficiency under operating conditions. These losses include: (1) heat of exhaust steam; (2) initial condensation; (3) re-evaporation; (4) incomplete expansion; (5) leakage; (6) radiation; (7) clearance volume; and (8) wiredrawing.

204. Exhaust Losses.—The largest loss of heat in both the ideal and the actual steam-engine cycle is the quantity of heat rejected by the engine and carried away by the exhaust steam. Cheap and poorly designed engines often discard more than 90 per cent of the total heat supplied. Other engines of better design and construction, together with the aid of auxiliary equipment, are able to reduce the amount of heat thus wasted. The exhaust losses are seldom less than 70 per cent of the total heat of the supplied steam, even with the highest grade of equipment operating under the most favorable conditions.

205. Initial Condensation and Re-evaporation.—Steam entering the cylinder of a counterflow engine strikes comparatively cool port and cylinder walls. The port and cylinder walls are cooled by the expanded exhaust steam, the temperature of which is reduced in the process of doing work. The incoming steam at high pressure and temperature immediately gives a part of its heat to the surrounding cool surfaces. The result is that a part of the saturated steam is condensed, or if the steam is superheated the steam temperature is reduced. Condensation of steam represents a heat loss and adds to the difficulty of lubricating the cylinder walls. Next to the heat losses existent in the exhaust, initial condensation is one of the largest losses occurring when saturated steam is used.

A part of the condensation may be re-evaporated later in the stroke as expansion takes place with diminishing pressure of the steam. When re-evaporation occurs the necessary heat is abstracted from the steam of the cylinder. This heat becomes unavailable to the engine and is lost in the exhaust. So long as the interior surfaces of the cylinder remain wet the transfer of heat through them to the outside is augmented.

Initial condensation is affected by the speed of the engine and the temperature range maintained within the cylinder. The slower the engine speed the greater is the amount of time available for the transfer of heat from the steam to the cylinder walls. The more that steam is expanded within a single cylinder, the greater the temperature range and the lower the temperature of the exhaust steam, cooling the walls of the steam ports and the cylinder. In general, shortening the period of steam admission, by earlier cut-off, improves the steam economy of an engine; the effect produced is more than enough to offset the increased initial condensation losses. However, for single-cylinder engines cut-off earlier than one-fourth of the stroke is not economical as the condensa-

tion losses exceed the gains secured by a greater number of expansions of the steam.

206. Incomplete Expansion.—Certain types of valves and valve gears cannot be adjusted to give an early economical cut-off without interference with the other events of the stroke, that is, release, compression, and admission. Under such conditions the number of steam expansions is limited and the economy and thermal efficiency of the engine are reduced.

207. Leakage Losses.—Leakage of steam through the engine valves into the cylinder during the expansion part of the stroke reduces the number of steam expansions within the cylinder. Leakage around the piston into the opposite end, or leakage through the valves into the exhausting end of the cylinder, increases the back pressure against which the piston works. The latter condition represents steam lost to the exhaust without the production of mechanical energy.

208. Radiation Losses.—Unless the cylinder walls and heads are well lagged with insulating materials, a considerable amount of heat will be lost through them by conduction, radiation, and convection. Losses due to the same causes also occur from the piston rod and valve rod. One of the effects of these heat losses is to increase the initial condensation of the engine. Although radiation losses cannot be eliminated entirely, they can be reduced materially by using proper precautions.

209. Clearance-volume Losses.—Clearance volume of an engine cannot be eliminated as it is impossible to provide steam and exhaust ports without volume. Furthermore, the piston must not strike the cylinder heads. A graphical representation of clearance is given by Fig. 173, where OG represents the engine clearance and GH the total piston displacement on the pressure-volume diagram. The total volume of the cylinder is equal to the piston displacement GH plus the clearance volume OG. The volume of steam within the cylinder at the point of cut-off B is equal to the sum of the clearance volume OG and the volume of piston displacement completed, GI. Therefore, the actual number of expansions occurring in the cylinder is

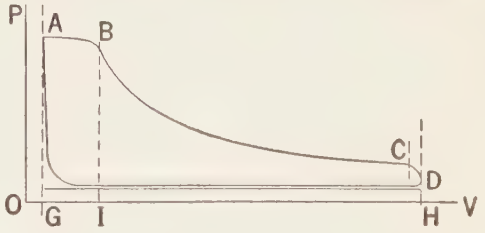


FIG. 173.—Effect of clearance volume.

$$r = \frac{OG + GH}{OG + GI}. \tag{123}$$

The weight of steam required in the cylinder at cut-off is that neces-

sary to fill the clearance at the pressure of point A and that necessary to fill the piston displacement at cut-off. The greater the magnitude of the clearance volume the greater is the weight of steam required to fill this unavoidable space. During exhaust the clearance is filled with low-pressure steam, and if the engine has no compression a greater weight of steam is required to fill the clearance at the pressure of point A. Therefore, compression is provided in the actual engine for two reasons: (1) reduction of the weight of steam required to fill the clearance volume; and (2) cushioning of the heavy reciprocating parts.

Steam, when compressed in the clearance space, incurs a rise in temperature due to the work done upon it. When the temperature of the compressed steam exceeds that of the clearance space walls, cooled by the exhaust steam, heat is transferred to the cooler surfaces. A thermal loss similar to that of initial condensation then takes place in the clearance space.

210. Wiredrawing.—Reductions of steam pressure due to the throttling action of valves as they open and close, and the restrictions of steam ports, affect the power output and economy of engines. The principal effect of wiredrawing is to reduce the availability of the heat energy by the pressure loss. Single-valve engines suffer greater losses due to wiredrawing than multivalve engines.

211. Methods of Increasing Engine Economy.—The following methods are commonly used to improve the economy and increase the thermal efficiency of the steam engine: (1) reduced exhaust or back pressure; (2) increased initial steam pressure; (3) superheating; (4) compounding; (5) steam jackets; (6) improved valve gears; (7) reheaters; and (8) single-flow cylinders.

212. Reduction of Exhaust Pressure The pressure of the expanded steam within an engine cylinder during the exhaust stroke is largely controlled by the pressure existing in the exhaust piping attached to the cylinders. Engines may operate with the exhaust pressure above atmospheric or less than atmospheric, depending upon the disposition of the exhaust steam. *Non-condensing* operation allows the steam to be wasted to the atmosphere or to be used at some pressure greater than atmospheric for heating or process work. *Condensing* operation necessitates the reduction of the steam to water within a closed chamber with a consequent reduction of the exhaust steam pressure. Condensing operation is accomplished and facilitated by the use of the equipment described in Chapter XVI.

Engines operating with fixed conditions of initial steam pressure, quality, and percentage of stroke at cut-off show a gain in the horsepower developed when the exhaust pressure is decreased. The theoretical gain in power, produced within a simple engine cylinder with

clearance, by reduction of the back pressure, is shown by the sectioned portion of Fig. 174. The effect is a greater horsepower for a given quantity of cylinder feed steam and a decrease in the water rate of the engine. If the engine is of a type in which the point of cut-off of the steam can be automatically varied, the amount of the cylinder feed is lessened for a given amount of power developed. The engine will show theoretically a decreased water rate and an increased thermal efficiency. An increase in the exhaust pressure produces the opposite effects.

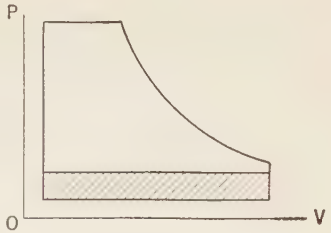


FIG. 174.—Effect of reduced exhaust pressure.

The foregoing theoretical considerations are partially true in the actual engine within certain limits. In single-cylinder engines of the counterflow type, the gain in steam economy is less as the back pressure is reduced. Leakage and initial condensation losses increase especially if the point of cut-off is made earlier in the stroke. Compound engines and simple single-flow engines are not affected so much by losses which tend to mitigate the beneficial effects of the reduced back pressure. In general, for each engine there is a critical back or exhaust pressure at which the engine gives its maximum economy.

Consideration must always be given to the heat consumption of the auxiliary equipment necessary to produce and maintain the reduced back pressure. The expenditure of heat energy in the condensing equipment together with the operating costs may render extreme reductions of the back pressure non-profitable. Unless an engine has specially designed exhaust ports, vacua of more than 22 in. of mercury in simple engines and 25-26 in. of mercury referred to a 30-in. barometer in multi-cylinder engines are not practicable. With the greater vacua largely increased cylinder volumes are necessary. Large cylinders offer mechanical problems as well as greater friction and initial condensation losses.

213. Increased Initial Pressure.—The use of high-initial-pressure steam permits more available energy per pound of steam supplied to the engine. The temperature range within the engine may be increased by the greater initial temperature of high-pressure saturated steam. Economy gains due to the use of increased initial steam pressures are not so limited in an engine as the gains secured by the reduction of exhaust pressures.

Modern commercial steam boilers are in operation under a pressure of 1800 lb. per sq. in., and steam engines are in use which are supplied with steam at the throttle under a pressure of 1440 lb. per sq. in. and a total temperature of 800 deg. F. The expansion of extremely high-

pressure steam should not be done in a single cylinder if all the benefits of the high initial pressure are to be secured. The maximum number of steam expansions should be secured, and the process should be completed in two or more cylinders. If wide temperature ranges are maintained in single cylinders the theoretical gains of the increased initial pressure are largely offset by the increased cylinder condensation and other heat losses.

Except in special cases, the maximum economies can usually be effected with pressures varying from 165 to 400 lb. per sq. in. gage. The economy gains with pressures above 400 lb. per sq. in. are of smaller magnitude and often do not warrant the increased initial and operating costs of equipment designed for high pressures. In general, good economy can be obtained with moderate steam pressures and high superheat.

214. Superheating.—In practical applications better results appear attributable to the use of superheated steam than theoretical considerations of the ideal steam-engine cycle indicate. The anomaly is due to the fact that initial condensation losses are not considered in the ideal cycle.

The use of superheated steam secures a marked decrease in the cylinder condensation and increased economy of the engine. Engines showing the largest amounts of initial condensation with saturated steam show the greatest savings with superheated steam. Under average conditions the steam consumption of an engine using saturated steam is reduced one per cent for each 10 deg. of superheat when superheated steam is used. Single-cylinder counterflow engines show the greatest gain in economy by the use of superheated steam.

Highly superheated steam may offer some disadvantages in the lubrication as well as warping and distortion of the rotating or sliding valves of certain types of engines. These troubles can be eliminated by the use of poppet valves for the admission of steam to the cylinder.

215. Compounding.—Cylinder condensation has been shown to be one of the important thermal losses of the steam engine. This loss is especially prone to occur in single-cylinder engines having early cut-offs and a large number of steam expansions in the one cylinder. Initial condensation losses can be reduced by dividing the number of expansions of the steam between two or more cylinders. In such an arrangement the pressure and temperature ranges, as well as clearance losses within the cylinders, are reduced and better steam economy is secured in the counterflow type of engine.

When the pressure and temperature ranges are divided between two cylinders the engine is designated as *compound*. Engines expanding steam successively in three cylinders are known as *triple-expansion*, and those using four successive cylinders for the steam expansion are called

quadruple-expansion engines. Engines which use multicylinders for expanding steam through a large pressure and temperature range are able to employ high initial steam pressures and the minimum feasible back pressures. Non-condensing compound engines are not desirable. The minimum initial steam pressures for engines having more than one cylinder for the progressive expansion of the steam are: compound, 125 lb.; triple-expansion, 200; and quadruple-expansion, 250 lb. per sq. in.

The advantages of compounding are: (1) greater range of steam expansion, (2) decreased cylinder condensation, (3) reduction of leakage and clearance losses, (4) possibility of more uniform power application to the crankshaft, and (5) lessened steam consumption per horsepower-hour. The foregoing beneficial effects are partially nullified by: (1) the increased initial cost, (2) multiplicity of parts, (3) increased floor space required, (4) decreased mechanical efficiency, and (5) increased heat losses by radiation.

216. Steam Jackets.—In some cases the cylinder walls and heads are made double and the space thus enclosed within them is filled with live steam at boiler pressure. Such arrangements are termed *steam jackets.* The function of steam jackets is to reduce the heat losses from the cylinder walls and heads and thus curtail cylinder condensation. In order to be effective the steam jackets must be free from air pockets and condensed steam, and must have unrestricted steam circulation. Steam-jacketed engines have shown improved economy in some cases and in others no improvement. Jackets are used to reduce the losses that superheat also tends to eliminate. The use of superheat is the better method of economy improvement, and jackets are not necessary or desirable where superheated steam is employed.

217. Improved Valve Gears.—Simple engines operating with valves and valve gears which do not permit early cut-off are not so economical as those in which the steam may be expanded a greater number of times. Furthermore, in such engines there is often a considerable amount of wiredrawing due to the restrictions of the steam passages and the sluggish action of the valve in opening and closing the ports. Valve gears designed to give early cut-off and quick action of the steam valves tend to improve the thermal efficiency of the unit. Also, by the use of more than one valve the length of the port passages may be shortened and the exhaust steam discharged from the cylinder through separate exhaust ports. The latter arrangement helps somewhat to reduce initial condensation losses.

218. Reheaters.—Multicylinder engines in which the steam is successively expanded in two or more cylinders usually have steam-storage spaces or receivers between the cylinders. These receivers may

be fitted with pipe coils through which high-pressure steam is circulated. The function of the heating coil within a receiver is to evaporate the moisture of the exhaust steam from the preceding cylinder and if possible to impart to the steam some superheat. Reheating is attempted in an effort to reduce the initial condensation losses in the cylinder following the receiver. If reheating is properly accomplished the effects in the low-pressure cylinders of multiple-expansion engines are the same as those of superheated steam in single-cylinder engines. Ample reheating eliminates the necessity for steam jackets for low-pressure cylinders.

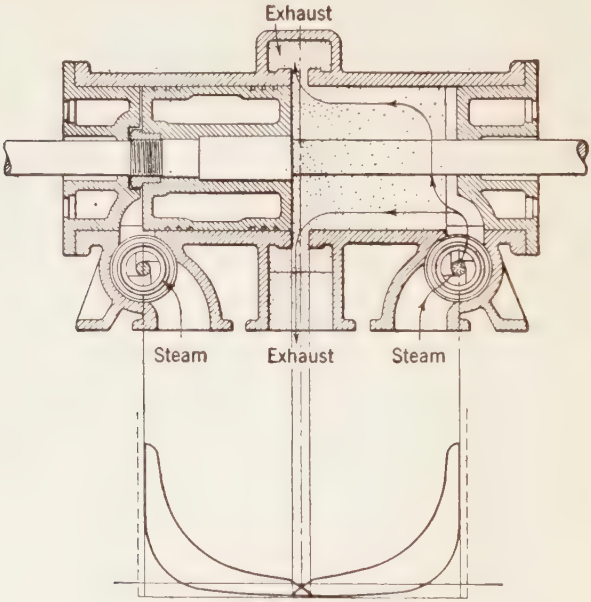


Fig. 175.—Diagrammatic section of a unaflow engine.

219. Single-flow Cylinders.—The counterflow type of engine operates at a disadvantage due to the fact that all the expanded and cooled steam is returned to the inlet end of the cylinder during exhaust. This cooled steam chills the cylinder walls and in many cases the port passages through which the hot live steam enters the cylinder. The result is that large initial condensation losses are produced. The single-direction or unaflow engine reduces this loss to a marked extent. Condensation is reduced by having separate inlet valves for the live steam and short port passages at each end of the cylinder, Fig. 175. The exhaust ports of the engine are located in the walls of the cylinder barrel midway between the cylinder ends. The engine piston serves as the

exhaust valve and alternately covers and uncovers the ports for exhaust from either end of the cylinder. The path of the steam in passing through the cylinder is in a single general direction, and thus cylinder condensation losses are reduced. Single-cylinder unaflow engines are able to show better steam economy than compound counter-flow multi-cylinder engines. One disadvantage incurred by these engines is the high-compression pressure developed, especially when operated non-condensing. This trouble is usually largely mitigated by the use of special auxiliary exhaust valves which are held open until near the end of the return stroke. This type of engine is described in more detail in Art. 225, Chapter XIV.

PROBLEMS

1. A steam engine used steam at 147.3 lb. per sq. in. gage throttle pressure and 97.5 per cent quality. Exhaust to the atmosphere; barometer, 29.92 in. of mercury. The engine output was 225 b.hp. at 90 per cent mechanical efficiency, and the total wet steam supplied per hour was 6970 lb. Find the water rate and the thermal efficiency based on i.hp.

2. A simple steam engine developed 40 b.hp. when supplied with steam at 135.5 lb. per sq. in. gage and 99 per cent quality. Engine exhaust was at 1.5 lb. per sq. in. gage, and the barometric pressure was 28.3 in. of mercury. The thermal efficiency of the engine based on b.hp. was 7.5 per cent. Find the water rate per b.hp-hr. and the total amount of wet steam used per hour.

3. A 200-hp. Corliss steam engine, at full load, was supplied with 20 lb. of dry saturated steam per b.hp-hr. when operating with an initial steam pressure of 165 lb. per sq. in. abs. and an exhaust pressure of 4 in. of mercury abs. Find the thermal efficiency of the engine.

4. A poppet-valve unaflow steam engine carrying a load of 450 b.hp. used 11 lb. of steam per b.hp-hr. when supplied with steam at 170 lb. per sq. in. gage and 137 deg. F. of superheat. Exhaust into a vacuum of 25.5 in. of mercury; barometric pressure, 29.66 in. of mercury. Mechanical efficiency 93 per cent. Find the thermal efficiency based on both b.hp. and i.hp.

5. A poppet-valve counter-flow engine gave the following test data: initial steam pressure, 138.3 lb. per sq. in. gage; barometer, 28.67 in. of mercury; initial steam temperature, 450 deg. F.; total power developed, 357 i.hp.; total steam used per hour, 7155 lb.; exhaust pressure, 0.75 lb. per sq. in. gage; and mechanical efficiency, 88 per cent. Find the water rate and thermal efficiency based on b.hp.

6. A 12-in. by 15-in. steam engine with a piston rod $1\frac{3}{4}$ in. in diameter gave the following test data: speed, 200 r.p.m.; initial steam pressure, 160 lb. per sq. in. abs.; exhaust pressure, 20 lb. per sq. in. abs.; m.e.p. head-end, 75.6 lb. sq. in.; m.e.p. crank-end, 77.9 lb. sq. in.; steam quality, 97.8 per cent; total steam used per hour, 4673 lb.; length of Prony brake arm, 63 in.; gross brake load, 510 lb.; brake tare, 56 lb. Find the water rate and thermal efficiency of the engine based on both the b.hp. and the i.hp.

7. A simple steam engine supplied with steam at 175 lb. per sq. in. abs. at the throttle used 40 lb. of dry saturated steam per b.hp-hr. when exhausting against an absolute pressure of 15 lb. per sq. in. What water rate and thermal efficiency may be expected if steam with a total temperature of 492 deg. F. is supplied to the engine? All other conditions of engine operation are to be unchanged.

8. A 500-kw. generator was direct-connected to a steam engine using steam at 225 lb. per sq. in. abs. and a total temperature of 450 deg. F. Exhaust was against a back pressure of 5.5 in. of mercury absolute. Mechanical efficiency of the engine 94 per cent and of the generator 90 per cent. Total steam used per hour at full load was 16,120 lb. Find the engine water rate and thermal efficiency based on i.hp.

9. How much would it be necessary to reduce the water rate of the engine of problem 8 to increase the thermal efficiency 2 per cent?

10. A 300-b.hp. engine operating at full load was used to drive a generator delivering 210-kw. output. Initial steam pressure, 145 lb. per sq. in. gage; steam quality, 98 per cent; exhaust into 24.5 in. of mercury vacuum; barometric pressure, 29.1 in. of mercury; and total wet steam used per hour, 7280 lb. Find the water rate and thermal efficiency of the engine.

11. The Rankine cycle efficiency of an engine was 22.5 per cent and the actual thermal efficiency 7.3 per cent. What was the engine efficiency?

12. A steam engine used 25 lb. of steam per b.hp-hr. when supplied with steam at 200 lb. per sq. in. abs. and a temperature of 425 deg. F. Engine exhaust pressure 15 lb. per sq. in. abs. Find the engine efficiency.

13. Calculate the engine efficiency from the engine data given in problem 4.

14. A 300-hp. Corliss steam engine operated under the following conditions: initial steam pressure 165 lb. per sq. in. abs.; steam quality, 99 per cent; exhaust pressure 4.5 in. of mercury abs.; water rate, 17 lb. dry steam per b.hp-hr. Find the engine efficiency.

15. A simple non-condensing steam engine used 30 lb. of dry steam per b.hp-hr. when supplied with steam at 125 lb. per sq. in. abs. and 99.5 per cent quality. Exhaust pressure 20 lb. per sq. in. abs. Find the engine efficiency.

CHAPTER XIV

MULTIVALVE AND MULTIEXPANSION ENGINES

220. Foreword.—Engines equipped with two or more valves have operating and other advantages that secure better thermal economy. Early cut-offs are permissible without interference with the proper distribution of the steam within the cylinder, as the valves may be adjusted independently. The action of the valve at cut-off may be

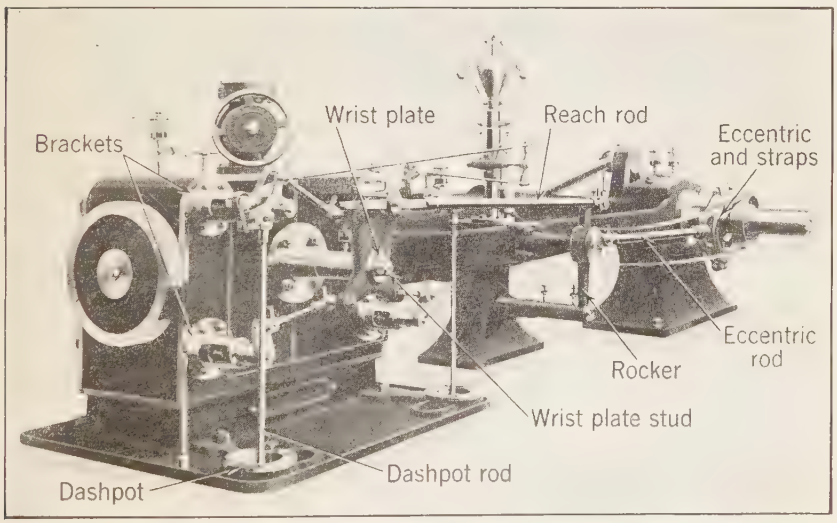


Fig. 176.—Murray Corliss standard steam engine.

made more rapid and wiredrawing losses due to the valve reduced. Port passages may be made shorter and the clearance of the engine reduced. The greater part of the gains in thermal efficiency secured are due to the reduction of initial condensation and wiredrawing losses.

221. The Corliss Engine.—In the middle of the nineteenth century George Corliss developed the four-valve type of engine named for him. This engine showed improved economy over anything in use prior to that time. Figure 176 illustrates a typical single-eccentric Corliss engine with a girder frame. The cylinder casting is made as shown by Fig. 177 and has four valve chambers. The two steam valves used for the admis-

sion and cut-off of the steam are located in cylindrical chambers at the top of the cylinder casting. The exhaust events of the cycle are controlled by cylindrical exhaust valves located at the bottom of the

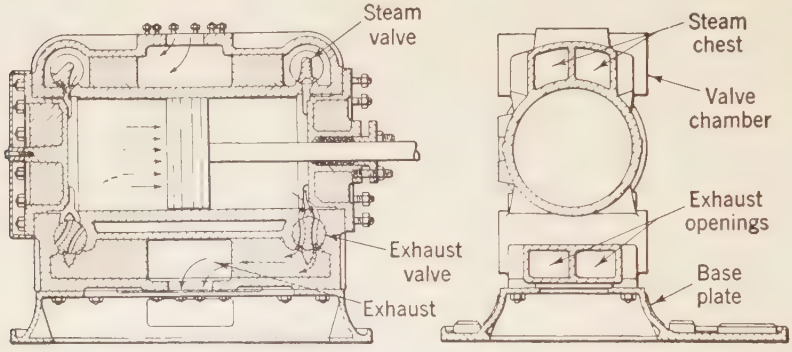


Fig. 177.—Murray Corliss engine cylinder with valves.

cylinder block. Very short ports lead from the valve chambers to the cylinder bore. The steam and exhaust valves are constructed as shown by Fig. 178, and are given a semi-rotary motion by the valve gear. The exhaust valves illustrated are double ported, and the steam valves are constructed to give the effects of double ports. Corliss valves are made with single, double, or triple ports.

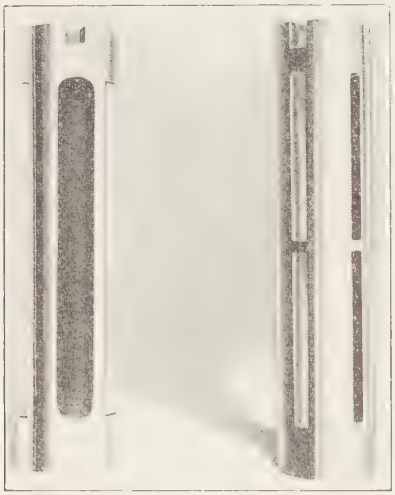


Fig. 178. Corliss steam and exhaust valves.

With the exception of the valves and the valve gear, the essential features of a Corliss engine are those common to many steam engines. The valves of the engine of Fig. 176 are actuated by a single eccentric mounted upon the engine shaft. The eccentric motion is transmitted by the eccentric straps and eccentric rod to a vertical rocker arm pivoted at its bottom end. Motion from the upper end of the rocker is carried by the reach rod to the wrist plate mounted upon a stud fastened to the side of the cylinder casting. The wrist plate is given an oscillating motion which is used to move the valves.

Each valve is rotated by a T-headed spindle which fits into the slot

in the end of the valve. Bearings for the valve spindles are located in the supporting steam brackets, and steam-tight packings are provided about the valve spindles to prevent leakage of steam from the valve chambers. Each valve spindle has an arm keyed rigidly to it. The levers attached to the steam valve spindles are steam arms, and those attached to the exhaust valve spindles are exhaust arms. Exhaust valve rods connect the exhaust arms and the wrist plate directly, thus the exhaust valves are opened and closed by the wrist plate. In the case of the steam valves the steam-valve rods driven by the wrist plate are attached to double arms, Fig. 179. The double arms are mounted upon the valve spindles, and are free to turn upon them. Each double arm carries a steam hook at the end opposite to the steam-valve rod

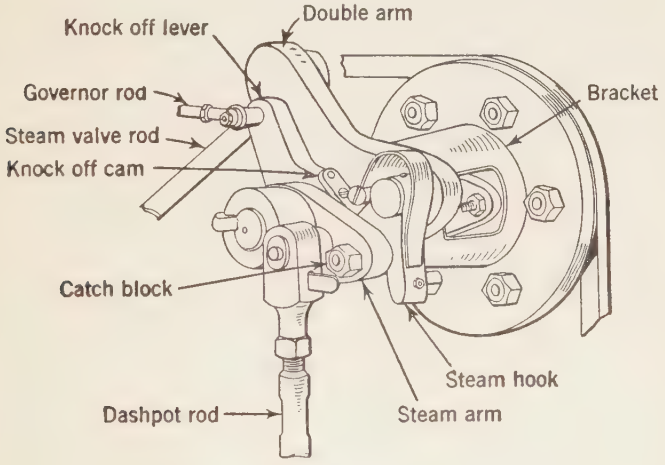


Fig. 179.—Murray Corliss steam-valve gear.

attachment. When the double arm is rotated to the proper position by the wrist plate the steam hook engages with a catch block fastened to the steam arm. Motion of the double arm causes the valve to be rotated to a position for admission of steam to the cylinder. Attached to the steam arms are rods which lift dash-pot plungers, Fig. 176. The dash-pot plungers are lifted against the pressure of the atmosphere as a partial vacuum is produced under the plunger when it is lifted. Admission of the steam into the cylinder continues until the steam hook comes in contact with the knock-off cam mounted upon the knock-off lever, Figs. 179 and 180. When this contact is made the steam hook is disengaged from the catch block, and the valves are quickly closed by the air pressure exerted on top of the dash-pot plungers. Thus very rapid and sharp cut-off is secured. The knock-off levers rotate about the

axes of the valve spindles, and the positions of the knock-off lever and the knock-off cam are directly under control of the governor.

Governing is effected by a loaded pendulum governor shown by Fig. 180. The governor action is transmitted by the long governor rod to the knock-off lever at the head end of the cylinder and by the short governor rod to the knock-off lever at the crank end of the cylinder. Cut-off governing is effected, and the speed regulation of the engine is generally good. Speed changes may be made by increasing or decreasing the amount of shot held in the governor weight, Fig. 180. Owing

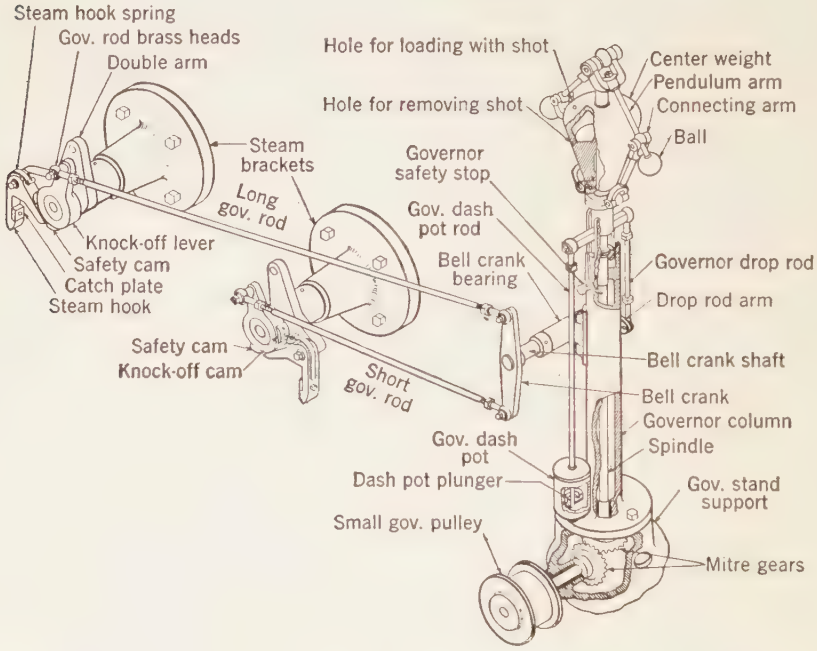


Fig. 180.—Murray Corliss engine governor.

to the fact that the point of cut-off is under governor control, the number of steam expansions in a single cylinder are variable, depending upon the load on the engine.

If the governor belt breaks, the speed of the engine is brought under control as the governor weights cease to revolve. This control is accomplished by the governor rods shifting the knock-off levers to a position where parts known as safety cams prevent the steam hooks from engaging with the catch blocks. When this action occurs the steam valves are not opened and the engine stops.

Because of the reduced clearance volume, reduced initial condensation

losses, and cut-off governing, the Corliss engine gives better heat economy than throttling-governed slide-valve engines. Typical indicator diagrams for a single-eccentric Corliss engine at full and varying loads are shown by Fig. 181*a*. The effects of the governor action may be seen by Fig. 181*b*.

222. Corliss Valve Gears.—The types of Corliss valve gears in use are: (1) single eccentric and double eccentric; (2) releasing; (3) non-releasing; and (4) long range.

The single-eccentric gear, of the engine of Fig. 176, cannot be used with cut-off much later than 3/8 of the stroke. If later cut-off is attempted with all valves actuated by a single wrist plate, the events of release and compression cannot be properly timed. In order to eliminate the interferences developed with late cut-offs in single-eccentric engines, Corliss engines are built with two eccentrics and two wrist plates. One eccentric with its wrist plate moves the steam valves, and the other with its wrist plate controls the exhaust valves. Engines equipped with two eccentrics may have cut-off as late as 7/8 of the stroke.

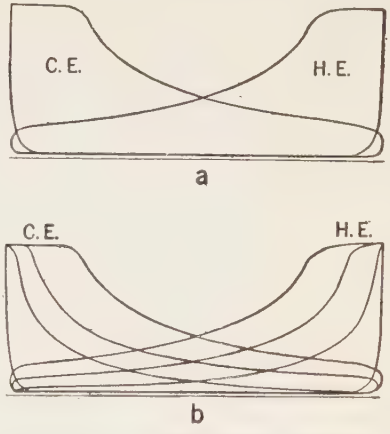


FIG. 181.—Corliss engine indicator diagrams.

The engine, Fig. 176, has a *releasing gear*, in that the steam valves are disengaged from the driving gear at cut-off, and are closed by the action of the dash-pots. Engines thus equipped are not suited to rotative speeds above 100 to 125 r.p.m., as the releasing gear and the dash-pots do not function properly with greater speeds.

Four-valve engines of the Corliss type suitable for higher rotative speeds are built with a *non-releasing gear* (Art. 223). Single wrist-plate engines can be given a *long range* of cut-off by the attachment of oscillating cams to the steam arms. These cams act to positively throw the steam hooks or latches out of engagement at the proper place in the stroke and thus secure cut-off where desired. The oscillating cams are moved to the proper positions by the governor acting through the governor rods. The cam motions are derived from a separate eccentric, and the point of cut-off is determined by the cut-off eccentric and the governor. The long-range gear permits higher rotative speeds, and cut-offs up to 0.8 of the stroke may be obtained by single wrist-plate engines.

223. Non-releasing Corliss Engine.—A section of a cylinder using Corliss valves and a non-releasing gear is illustrated by Fig. 182. The steam valves shown are double ported, although they may be made single or triple ported, depending upon the cylinder size.

The steam valves are actuated by links and levers enclosed within oil-tight housings, Fig. 183. The motion imparted to the steam valves is almost identical with that obtained from a releasing gear with the added advantage that higher rotative speeds are permissible. The steam valves are opened and closed at the proper times and are held stationary during the remainder of the time. Small movements of the steam valves are required, and they are balanced while opening and are

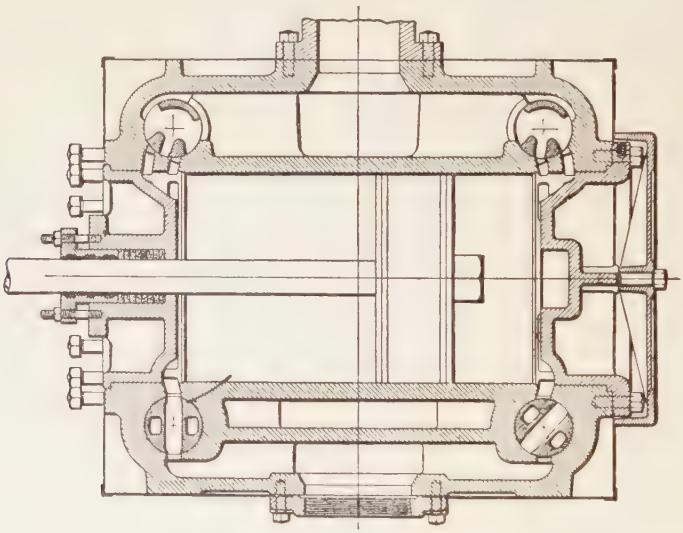


FIG. 182.—Erie Ball non-releasing Corliss engine cylinder.

nearly balanced during the period of closure. The exhaust valves are driven from a wrist plate as is customary with Corliss engines.

Governing is effected by the use of a flywheel governor which changes the position of the shifting eccentric used to operate the steam valves.

224. Poppet-valve Engine.—Poppet-valve engines are a European development adapted to the use of highly superheated steam. The longitudinal section of a poppet-valve engine is shown by Fig. 184. This engine has two inlet and two exhaust valves, all of which are operated by means of eccentrics and rods from a lay-shaft driven by the bevel gears, Fig. 185. The poppet valves are of the double-seat type. A detail of a steam inlet valve and the operating gear are shown by Fig. 186. The exhaust valves and their gear are similar.

The engine is governed by a combination centrifugal-inertia governor, mounted upon and driven by the lay-shaft. Speed variations of the engine cause centrifugal weights and an inertia ring within the governor housing to be shifted. The movement of the component parts within the governor housing causes the throw of the slotted eccentrics operating the steam valves to be changed. In this manner the point of cut-off is varied within the engine cylinder to suit the load conditions. Speed changes of the engine may be effected, while the engine is in operation,

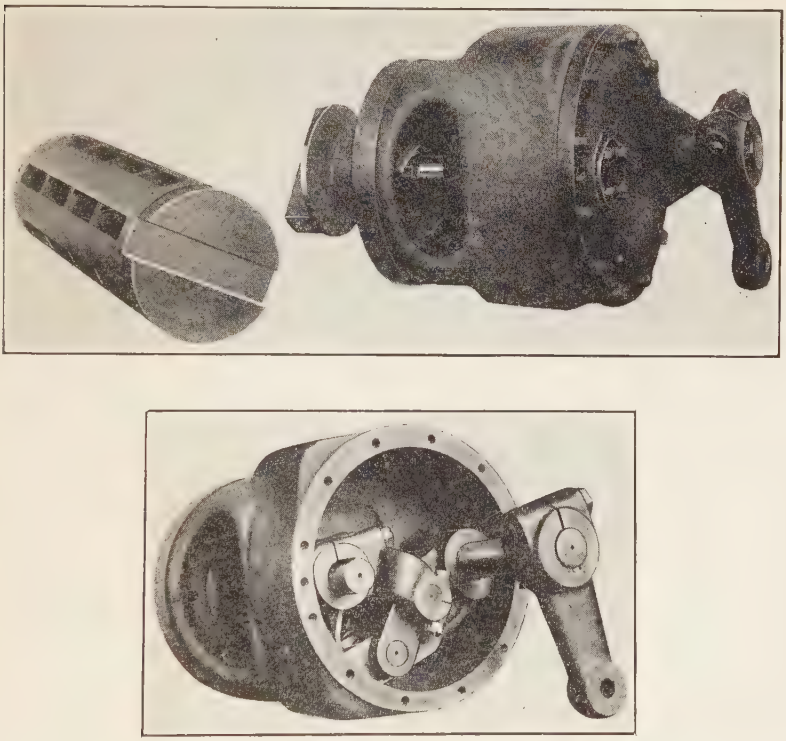


Fig. 183.—Armstrong non-releasing valve gear.

by adjusting the tension of the governor spring. This adjustment is accomplished by the aid of a small hand wheel at the end of the lay-shaft.

The valve spindle movements are derived from the eccentric rods by the action of a bell crank cam. The cams bear on rollers until the valves are seated by the action of the valve springs. The cam contacts are disengaged as the valves seat, noise is eliminated and the engine may be operated at any reasonable speed.

Poppet valves permit the use of highly superheated steam as diffi-

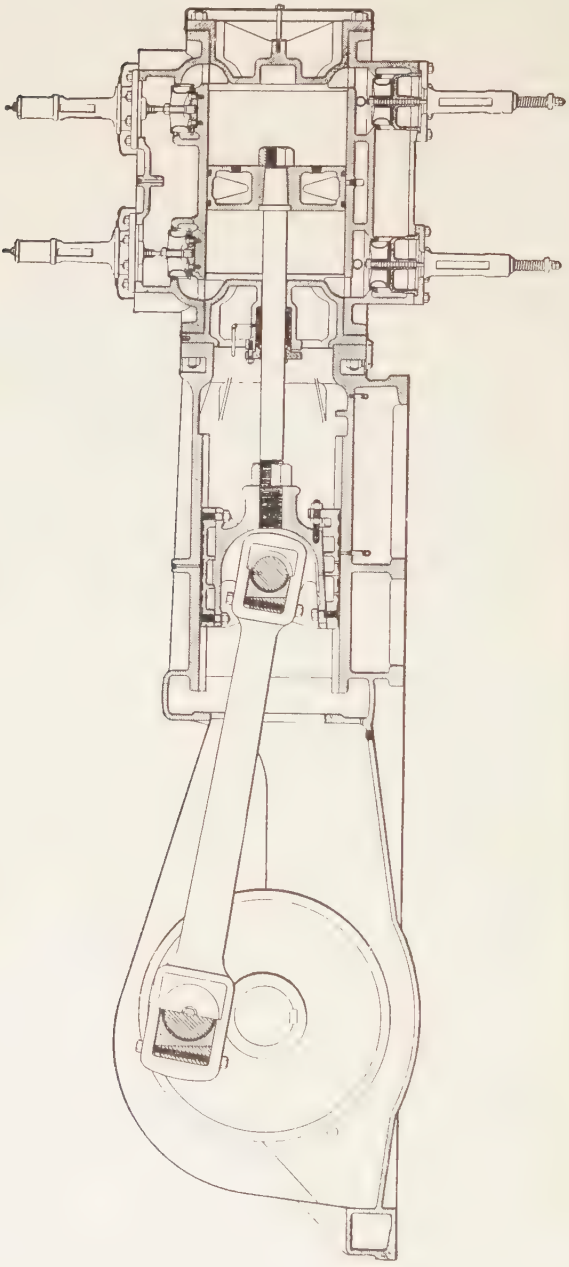


FIG. 184.—Longitudinal section of an Erie City Lentz poppet-valve steam engine.

culties of lubrication and warping of valves are eliminated. The use of the highly superheated steam, together with reduced clearances, etc.,

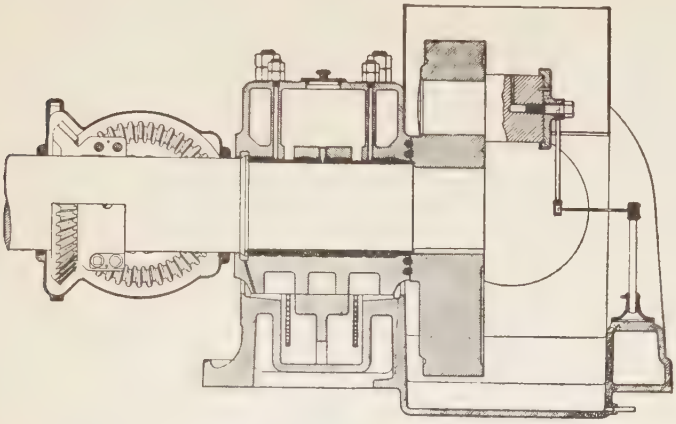


FIG. 185.—Section of the main bearing of a Lentz poppet-valve steam engine.

reduce the thermal losses within the engine and good operating economy results. These engines are built both in simple and compound units.

225. Unaflow Engine.—The single-flow principle, as well as the

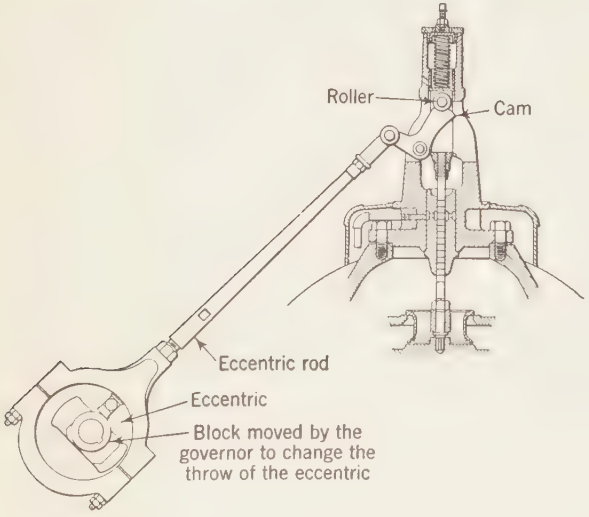


FIG. 186.—Steam-inlet valve-gear of a Lentz poppet-valve engine.

benefits derived from its use, were explained by Art. 219. The best results due to the application of the single-flow principle are obtained from condensing engines having a single valve at each end of the cylinder,

as the example of Fig. 187. Condensing operation is desirable for all unaflow engines, as ordinarily compression of steam occurs during about 90 per cent of the return stroke. In condensing operation the compres-

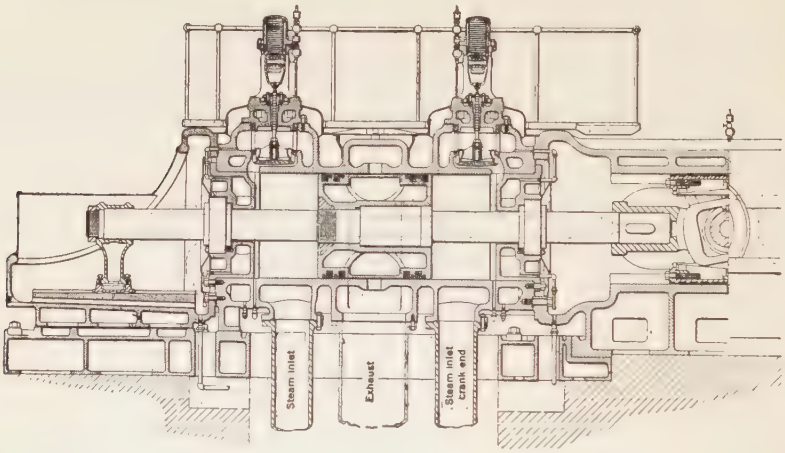


Fig. 187.—Sectional elevation of a Mesta unaflow engine cylinder. (Stumpf patents.)

sion pressure may rise to that of the initial steam, but in non-condensing operation the compression pressure, unless relieved in some manner, will exceed the initial steam pressure. Condensing unaflow engines,

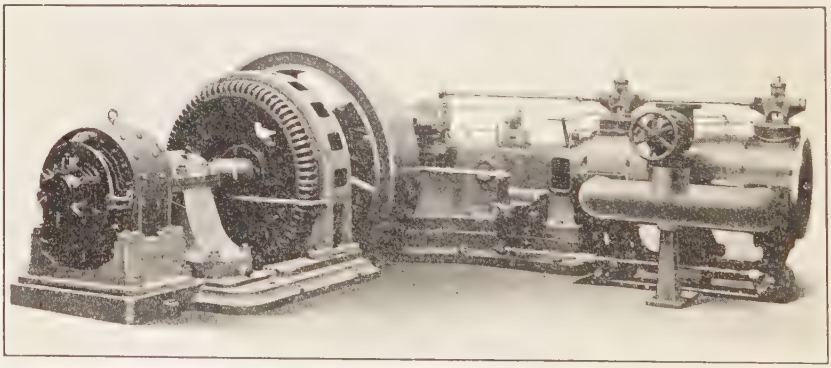


Fig. 188.—Elliott simple unaflow engine, with auxiliary exhaust valves.

operating with exhaust pressures considerably below atmospheric pressure, have a reduced weight of steam entrapped within the cylinder, when the piston covers the regular exhaust ports. Consequently, as the weight of steam compressed is not large, the pressures of compression do

not become excessive. Engines operating non-condensing have a greater weight of steam held within the cylinder during compression, and the pressures become excessive with a loss of engine power and steam economy. Non-condensing engines are often fitted with auxiliary clearances, steam ports, or automatically or mechanically operated relief valves to reduce the compression pressure. These devices permit the development of more power than could be obtained without their aid in non-condensing operation, but the use of the expedients to relieve compression partially nullify the thermal gains to be secured by application of the unaflow principle. However, the non-condensing unaflow engine

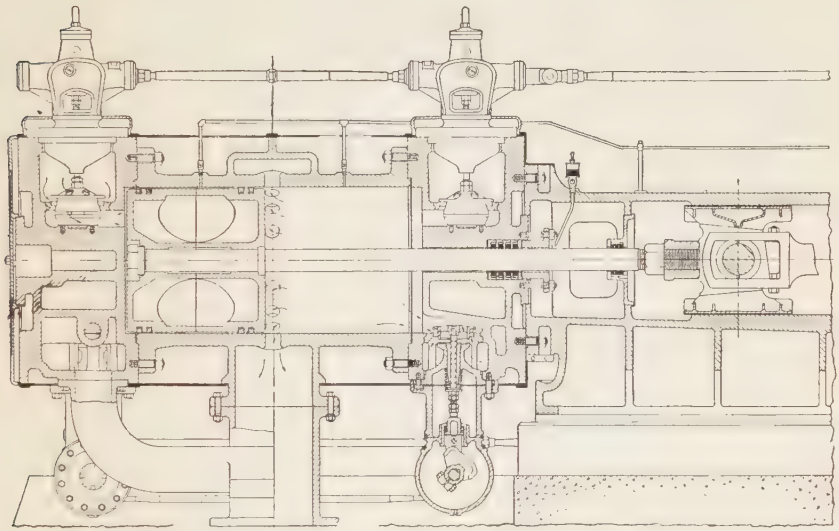


Fig. 189.—Sectional elevation of an Elliott unaflow engine, showing an auxiliary exhaust-valve and the valve-operating mechanism.

is able to show materially lower water rates than counter-flow engines operating under comparable conditions.

Figure 188 shows a view of the flywheel side of a unaflow engine with auxiliary exhaust valves. The valve arrangement of the engine is shown by Fig. 189. Two poppet inlet-valves are provided for the admission of steam at the top of the cylinder. The usual exhaust ports, opened and closed by the engine piston, are located at the middle of the cylinder barrel. A part of the steam entrapped in the cylinder after the regular exhaust ports are closed by the piston is released through the mechanically operated poppet exhaust valves at the bottom of the cylinder. With the exception of the two exhaust valves, the construction is similar to that of any two-valve unaflow engine.

Figure 190 illustrates a cylinder relief valve which is used by Elliott unaflow engines built for condensing operation. This valve is used when the vacuum fails and the compression pressures become excessive. The small inside valve is hand-operated and serves as a cylinder drain. The larger valve is a spring-loaded valve which can be adjusted by an outside nut. When the compression pressure becomes excessive the spring-loaded valve opens under the pressure and permits compressed steam to flow into the exhaust. When non-condensing operation is necessary the large outside valve may be opened by means of the hand wheel, and a port uncovered between the cylinder and a clearance pocket.

226. Economy of the Unaflow Engine.—The improved thermal efficiency of the unaflow engine is secured primarily by the reduction of condensation losses, through the use of the single direction of steam

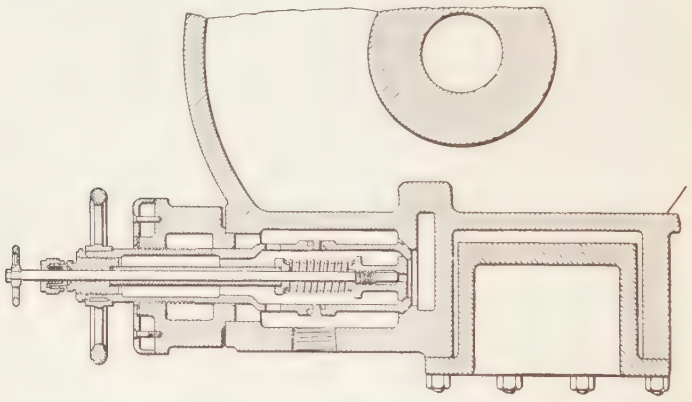


Fig. 190.—Elliott unaflow engine relief valve, for use in non-condensing operation.

flow and the use of jacketed cylinder heads and barrels. Turbulent and cooled steam does not reduce the temperature of the cylinder walls and ports at the inlet end of the cylinder, and a minimum temperature difference is always maintained between that of the steam and that of the confining walls. The exhaust passages around the cylinder barrel are filled with steam which, in a way, serves as a cylinder jacket. The cylinder clearances are reduced as well as steam leakage through the valves. The engine is adapted to high initial steam pressures and temperatures, and high rotative speeds, and is not handicapped by difficulties of valve lubrication and valve warping.

Unaflow engines have smaller variations of the amount of steam consumed per horsepower-hour with variation of loads than compound engines show, Fig. 191. As a matter of fact, most compound engines show good economy at one load only. Comparative indicator diagrams of a single-cylinder unaflow engine and a compound engine, operating

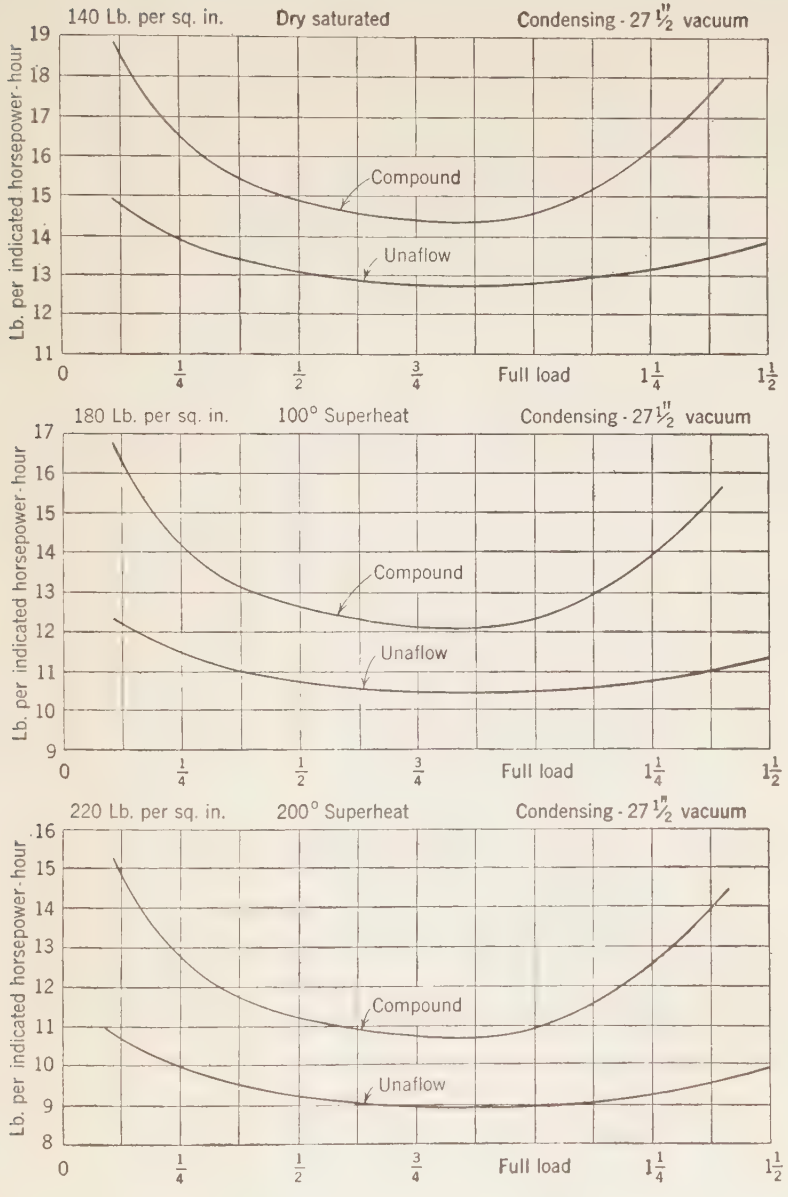
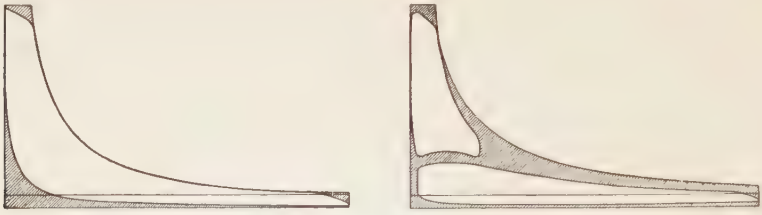


FIG. 191.—Steam consumption curves. Mesta single-cylinder unaflow, and a compound counterflow engine.

under identical conditions, are shown by Fig. 192. The diagrams of the two cylinders of the compound engine have been combined into a single one, and the theoretical diagrams for both the unaflow and the compound engine have been superimposed upon the unaflow diagram and the combined diagrams of the compound engine. The cross-sectioned areas represent the lost work areas in each case.



Unaflow. Compound counterflow.

FIG. 192.—Comparison of unaflow and compound-engine indicator diagrams.

Unaflow engines are more economical in many cases than small turbines operating under comparable conditions, and in recent years have largely superseded the compound steam engine where variable loads imposed in engine operation occur.

227. Arrangement of Compound Engines.—Engines with two or more cylinders for the successive expansion of steam are built so that the classifications of Art. 166, Chapter XI, are applicable. In addition

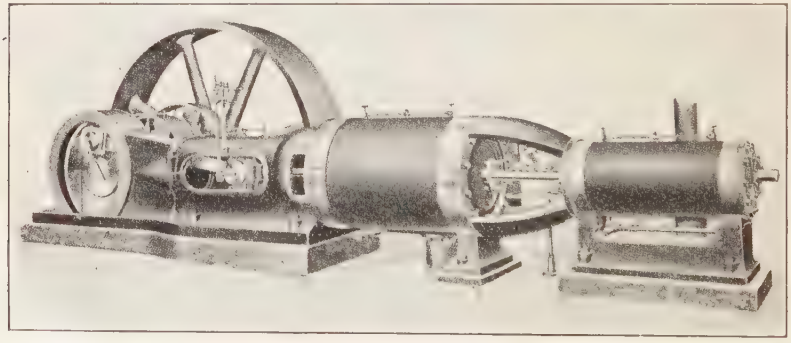


FIG. 193.—Buckeye tandem-compound steam engine.

to the groupings there given, further differentiations of the cylinder locations may be made: (1) tandem-compound, cylinder longitudinal axes on the same center line; (2) cross-compound, cylinder longitudinal axes on parallel center lines—engine cranks at opposite ends of the crankshaft; (3) duplex-compound, cylinder longitudinal axes parallel—both cylinders served by the same crankshaft but with separate crossheads and connecting rods; and (4) angle-compound, cylinder longi-

tudinal axes at right angles. The tandem and cross-compound arrangements are those most commonly used.

228. Tandem-compound Engines.—Engines with the cylinders in tandem have the pistons of the two cylinders mounted upon a single piston rod, Fig. 193. The low-pressure cylinder is usually placed nearest the crankshaft. This location restricts access to the cylinder. Tandem-compound engines require a smaller amount of floor space than those of the cross-compound type.

Tandem-compound engines may be built with valves controlling cut-off of the steam in the high-pressure cylinder only, or with valves controlling cut-off in both the high- and low-pressure cylinders. Where a cut-off valve is provided for the high-pressure cylinder only, the flow of steam is as follows: During admission and expansion of steam in the head end of the high-pressure cylinder, exhaust steam flows from the crank end of the high-pressure cylinder into the head end of the low-pressure cylinder. During admission and expansion of the steam in the crank end of the high-pressure cylinder, exhaust steam flows from the head end of the high-pressure cylinder into the crank end of the low-pressure cylinder. In each case, cut-off takes place in the low-pressure cylinder at the end of the stroke, and disposal of the exhaust from the low-pressure cylinder is made in the usual manner. Connecting pipes are provided between the proper cylinder ends. Such an engine is of the "Woolf" type. When valves are provided to give cut-off in both the high- and low-pressure cylinders the use of a *receiver* is required.

229. Engine Receivers.—Storage space for exhaust steam from the high-pressure cylinder is necessary, and is secured by the use of receivers for the following conditions: (1) When the exhaust of steam from the high-pressure cylinder cannot pass continuously to the low-pressure cylinder. (2) Engines having the low-pressure crank ahead of the high-pressure crank and requiring admission of steam to the low-pressure cylinder, when the high-pressure cylinder is not exhausting. Receivers may consist merely of enlarged pipes between the cylinders or they may be built of steel plates riveted together. When reheating of the steam is desired between cylinders, coils for high-pressure steam are inserted within the receiver.

230. Cross-compound Engine.—An engine of this type usually has overhung cranks at the ends of the crankshaft with the engine flywheel mounted between the main bearings of the engine. Ordinarily, the low-pressure cylinder crank is set 90 deg. ahead of the high-pressure crank, in the direction of rotation. This arrangement permits a more uniform turning effort on the crankshaft, and necessitates the use of a receiver. A typical cross-compound engine having high- and low-pressure cylinders

of the four-valve type is illustrated by Fig. 194. Engines of this type require considerable floor space, but the parts may be made lighter.

231. Compound Engine Cylinder Ratio.—The ratio of the piston displacement of the low-pressure cylinder to that of the high-pressure cylinder is the cylinder ratio. For any engine the ratio is

$$R = \frac{\frac{\pi}{4}D_2^2 l_2}{\frac{\pi}{4}D_1^2 l_1} = \frac{D_2^2 l_2}{D_1^2 l_1}, \tag{124}$$

in which D_1 = diameter of high-pressure cylinder, in.
D_2 = diameter of low-pressure cylinder, in.
l_1 = length of stroke high-pressure cylinder, in.
l_2 = length of stroke low-pressure cylinder, in.

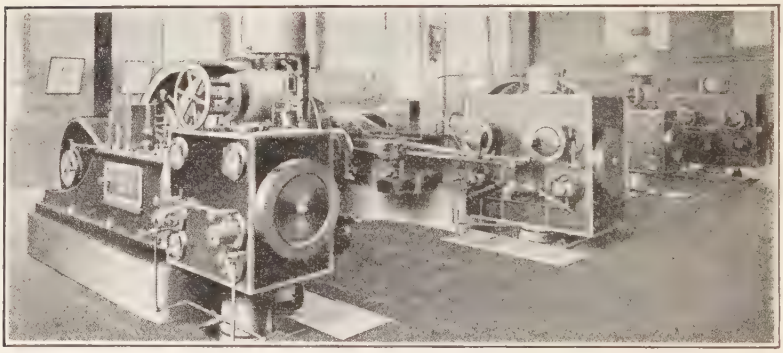


FIG. 194.—Erie Ball cross-compound engine with non-detaching valve gear.

When the strokes of the high- and the low-pressure cylinders are equal, as often is the case

$$R = \frac{D_2^2}{D_1^2}. \tag{125}$$

The cylinder ratio should be such that a small drop in steam pressure occurs between cylinders when the power developed by them is equally divided.

232. Ratio of Expansion.—The ratio of the total volume of the low-pressure cylinder to that of the high-pressure cylinder at the point of cut-off is the actual expansion ratio r. In each cylinder the volume under consideration is the piston displacement plus the clearance volume of the cylinder end under consideration.

233. Power of a Compound Engine.—Engines operating without bleeding of steam from the receivers between cylinders discharge in the

low-pressure exhaust practically all the steam supplied to the high-pressure cylinder. The calculation of the theoretical power of a compound engine is made on the basis that all the work may be done in a single cylinder having the dimensions of the low-pressure cylinder. The number of steam expansions in the low-pressure cylinder is considered as being the same as the total of the expansions of both the high- and the low-pressure cylinders. On this basis the theoretical mean effective pressure of an equivalent simple engine may be calculated by the use of equation 111, Chapter XI. As the probable m.e.p. is always less than the theoretical m.e.p., it is necessary to apply a diagram factor to the m.e.p. thus found. Diagram factors for compound engines range from 0.5 to 0.8, depending upon the type of the engine and the valve gear used. Any type of valve and valve gear may be used, such as slide, Corliss, or poppet. The probable m.e.p. having been found, the probable power of the engine for each cylinder end may be calculated by the use of equation 112, Chapter XI. The actual power of a compound or multiexpansion engine is the sum of the powers developed simultaneously in all the cylinders.

234. Governing.—Compound engines, except some used for driving air compressors, are governed by varying the point of cut-off. The governors used are of the centrifugal or inertia shaft type, or pendulum-loaded governors for engines with Corliss releasing valve-gears.

Cut-off governing may change only the point of cut-off in the high-pressure cylinder or the point of cut-off may be varied in both the high-pressure and the low-pressure cylinders. The total power developed by a compound engine is dependent upon the point of cut-off in the high-pressure cylinder. The distribution of work or power between the cylinders is adjusted by the point of cut-off in the low-pressure cylinder. Making the cut-off earlier in the low-pressure cylinder increases the power of the low-pressure cylinder, and decreases the power of the high-pressure cylinder. The effect is caused by the increased receiver pressure, which causes the high-pressure piston to work against a greater back pressure and the low-pressure cylinder to work with a higher initial pressure. Late cut-off in the low-pressure cylinder produces the opposite effect, as more work is done in the high-pressure cylinder and less in the low-pressure cylinder.

235. Triple- and Quadruple-expansion Engines.—Engines expanding steam in four successive cylinders or stages are seldom used at the present time. Triple-expansion engines expand the steam in three or more cylinders. It is not uncommon to find triple-expansion engines fitted with five cylinders: one for the partial expansion of high-pressure steam, two intermediate cylinders, and two low-pressure cylinders.

The common applications of triple-expansion engines in recent years

have been only to the propulsion of ships and operation of pumping units for water supplies. The recent tendency has been to discard the triple-expansion engine in both of these services. Modern ships are driven either by turbine generators through electric motors, by turbines through speed-reduction gears, or by internal-combustion engines of the Diesel type. Although the triple-expansion engine has shown remarkable economy in pumping plants, the general tendency is to use centrifugal pumps, which are driven by electric motors. Although the centrifugal pump thus driven does not show as good performance in the matter of economy, the amount of floor space required and the greater simplicity of the pumping unit are factors favoring the discard of the triple-expansion engine.

A notable and interesting installation of triple-expansion steam engines is in the plant of the Philip Carey Company, at Lockland, Ohio, where two 6060-i.hp. engines are each direct-connected to a 4700-kva., 2300-volt generator operating at a speed of 225 r.p.m. These engines are vertical units with five cylinders and receive steam at the throttle under a pressure of 1440 lb. per sq. in. and a temperature of 800 deg. F.

The stroke of each engine is $25\frac{5}{8}$ in. Each engine has two high-pressure single-acting cylinders, $14\,\frac{3}{16}$ in. in diameter; two immediate single-acting cylinders, $21\frac{1}{4}$ in. in diameter; and one low-pressure double-acting cylinder, $33\frac{1}{2}$ in. in diameter. The steam is reheated to 525 deg. F. between the intermediate cylinders and the low-pressure cylinder. The latter exhausts the steam at 60 lb. per sq. in. to an industrial evaporator.

CHAPTER XV

STEAM TURBINES

236. Development and Application.—At the beginning of the twentieth century the steam turbine began to be applied to commercial installations and today has supplanted the steam engine for electric generating stations. The steam turbine can be built in units of very much larger capacity than is practicable with reciprocating engines. However, the reciprocating engine is being used for certain classes of service such as small industrial plants, piston compressors, belt drives, hoisting engines, etc. The development of the turbine has been remarkably rapid, but at the same time improvements in other central station equipment have been made, especially in boiler-plant construction and operation, condensing apparatus, and electric generator design. Single-cylinder units have been built in various sizes ranging from 1 kw. to large turbine-generators of 80,000 kw. rated capacity. One manufacturer has expressed willingness to build single-cylinder turbines for operation at 1800 r.p.m. and steam conditions of 650 lb. gage and 825 deg. F. at the throttle for a capacity of 100,000 kw. A multiple-cylinder unit developing 208,000 kw. has been installed; one manufacturer has offered to build a multicylinder unit of 360,000 kw. The vertical-compound, also called steeple-compound, arrangement is extremely economical of floor space. In 1933, two 25,000-kw., two 50,000-kw., and one 110,000-kw. turbines of this type were in operation at 1200 lb. gage pressure. The high-pressure turbine and its generator are, in the four smaller units, mounted on top of the generator for the low-pressure turbines. In the larger unit a 55,000-kw. high-pressure turbine and its generator are mounted directly on top of the 55,000-kw. low-pressure turbine and its generator.

The steam in a turbine is continuously directed from nozzles or guide passages against semicircular blades (or buckets) uniformly spaced on the periphery of an enclosed rotor. The reciprocating steam engine utilizes the available energy of the steam by the action of a static pressure against the piston. In a steam turbine the energy transformation is also accomplished by a pressure action on the blades of the rotor, but the pressure is dynamical, being due to a change of momentum of the steam in its passage through the blade channels. As expansion pro-

ceeds in the steam engine, the heat energy is directly transformed into work on the piston. In the turbine there is a double transformation of energy; heat is first converted into kinetic energy by the expansion of the steam in the nozzles; this kinetic energy is then transformed into mechanical work on the rotor blades.

Compared with reciprocating engines of the same capacity and operating conditions, steam turbines occupy less floor space, require smaller and cheaper foundations, deliver oil-free exhaust steam, operate with highly superheated steam without lubrication difficulties, and, in general, show about the same heat economy and thermal efficiency. Other advantages are uniform velocity of rotation, higher rotative speeds, smaller initial cost, and the fact that less attendance is required. The steam turbine is primarily adapted to the driving of high-speed machinery such as electric generators, centrifugal pumps, rotary compressors, and fans, but is not applicable to direct drives where relatively low speeds are desired or where a large starting torque is necessary. The high speed of the turbine may be reduced by gears, but this arrangement increases the size, weight, and cost of the unit, and decreases the mechanical efficiency.

237. Classification of Steam Turbines.—Like reciprocating engines, steam turbines may be classified under the headings of *simple* and *compound*. A more general classification, in accordance with the conditions of operation of the steam on the rotor blades, is: (1) impulse, (2) pure reaction, and (3) reaction. However, in all important types of steam turbines, the blades are moved by both the impulse and the reaction of impinging steam jets issuing from nozzles or guide passages.

In the impulse turbine, Figs. 195 and 198, steam expands in a nozzle until the pressure reaches the pressure of the region in which the turbine wheel rotates. The steam jet issuing from the nozzle is then directed against the blades of the turbine wheel, and the *impulse* of the jet produces rotation of the turbine wheel. With the impulse type of turbine only a part of the rotor passages is filled with steam at any instant, even if several nozzles are used.

With the reaction type of turbine, steam flows through guide passages in a stationary ring (stator) *S*, Fig. 196, and then through passages in the circumference of the rotor *R*. These passages are filled with steam throughout the entire circumference; the steam is partially expanded in the stator, and the blades are so made that part of the expansion occurs in the rotor. In the *pure-reaction* turbine there are no stationary nozzle passages, and the expansion takes place wholly in nozzle-shaped passages at the circumference of the rotor. In the *reaction* type, Fig. 196, expansion takes place in both fixed and moving nozzle passages, and the pressure at the entrance to each rotating blade

is therefore greater than at the exit. The term reaction employed to denote this latter type of turbine is not strictly correct, as both impulse and reaction forces are present. However, the term reaction has come into general use and will be used in this chapter to denote this type of steam turbine.

238. Nozzle Design.—In the preceding articles the statement was made that heat is converted into kinetic energy when steam flows through a nozzle. An ideal impulse turbine could absorb all the kinetic energy of the steam jet and make this energy available in the form of useful power at the turbine shaft. The ideal turbine is, therefore, of interest in studying the velocity which a jet will acquire, the behavior of the steam, and the dimensions of the nozzle. To simplify matters, the assumption will be made that the steam neither receives heat from nor gives heat to other bodies during its passage through the (frictionless) nozzle, in other words, that the jet is formed under *adiabatic* conditions. Assume steam flowing through a nozzle of any form, from a region of relatively high pressure to one of low pressure. Each element of the stream expands, and the work done in expanding gives energy of motion to the element in front of it. The whole stream, therefore, acquires velocity in the process and also increases in volume. Let p_1 and p_2, Fig. 197, be imaginary partitions, across which the steam flows, taken at right angles to the direction of the stream lines. Let p_1 be the pressure of the steam at the entrance to the nozzle, V_1 the velocity, and v_1 the volume of one pound of steam entering the nozzle. Similarly, let p_2, V_2, and v_2 be the pressure, velocity, and volume of

Fig. 195.—Impulse turbine.

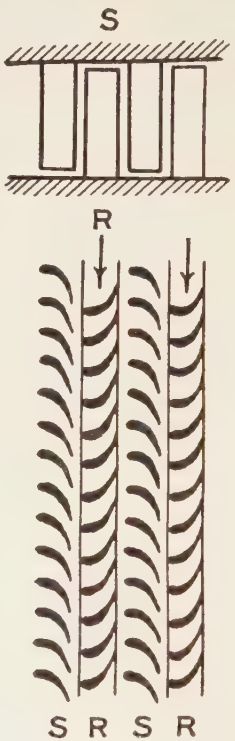


Fig. 196.—Reaction turbine.

one pound of steam at exit of the nozzle. As a result of the expansion during the drop in pressure from p_1 to p_2, the velocity increases from V_1 to V_2, and the increase of kinetic energy is equal to $\frac{V_2^2 - V_1^2}{2g}$ per pound of steam.

Each pound of steam that enters the space between p_1 and p_2 has work done upon it, amounting to p_1v_1, by the steam behind. In passing out of this space at p_2, the steam does work amounting to p_2v_2 on the steam in front of the jet. As the steam flows from p_1 to p_2, internal energy amounting to $(U_1 - U_2)$ is lost. Hence, by the principle of the conservation of energy, since by assumption no heat is taken in or given out,

$$\frac{V_2^2 - V_1^2}{2g} = (U_1 - U_2) + (p_1v_1 - p_2v_2). \tag{126}$$

But $(U_1 + p_1v_1)$ is h_1, the total heat at p_1; $(U_2 + p_2v_2)$ is h_2, the total heat at p_2; and the above equation may be written

$$\frac{V_2^2 - V_1^2}{2g} = J(h_1 - h_2). \tag{127}$$

The gain in kinetic energy is, therefore, equal to the loss of total heat, or what is commonly called the "heat drop." In many practical cases the initial velocity is negligibly small, so that

$$\frac{V^2}{2g} = J(h_1 - h_2), \tag{128}$$

where V is the velocity in feet per second acquired as a consequence of the heat drop. Since $g = 32.17$,

$$V^2 = 778.6 \times 64.34(h_1 - h_2),$$

$$V = 223.8\sqrt{h_1 - h_2}. \tag{129}$$

The cross-sectional area of a turbine nozzle at any point n may be computed by

$$A_n = \frac{Wv_n}{V_n}, \tag{130}$$

where A_n = area, sq. ft.
W = weight of steam flowing, lb. per sec.
v_n = specific volume of steam at pressure p_n, cu. ft. per lb.
V_n = velocity of steam at point n, ft. per sec., as determined from equation 129.

Thus the area of the nozzle at any point depends upon the weight, volume, and velocity of steam flowing. The assumption is that the flow is adiabatic and frictionless, hence the entropy of the steam at any point in the nozzle is equal to the initial entropy. The design of a turbine nozzle consists simply in choosing sections so that the desired amount of steam may flow with the desired pressure drop, as the velocity is determined by that pressure drop. This is done by working in terms of one pound of steam, since equations, steam tables, and charts generally give values for one pound. The cross-sectional area of a nozzle at any point for passing a number of pounds of steam may be determined by multiplying the area required for one pound by the number of pounds flowing.

The shape of the entrance end of a nozzle is generally rounded by some convenient curve, as shown in Fig. 197. Experiments and mathematical determinations indicate that the steam pressure p_0 at the throat of a nozzle falls to about 0.58 of the initial absolute pressure for saturated steam, and to 0.55 for superheated steam. For back pressures less than 0.58 of the initial, the nozzle must first converge from inlet to throat and then diverge from throat to exit in order to obtain a maximum velocity and provide for the increase in volume during expansion of the steam. Figure 197 shows a section through a theoretically proportioned expanding nozzle, in which $p_0 = 0.58p_1$ and $p_2 < 0.58p_1$. To prevent friction, eddies, and velocity losses, the divergence of the walls of non-condensing nozzles is about 3 deg. from the axis of the nozzle. Condensing nozzles for high vacua may have a divergence of as much as 8 deg. The total angle of divergence, therefore, varies from 6 to 15 deg. with an average value of 10 deg. For an angle of divergence of about 10 deg., the length L of the nozzle may be approximated by the equation

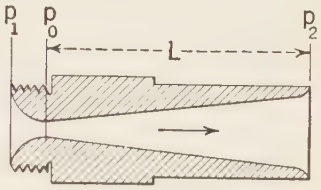


Fig. 197.—Form of turbine nozzle when back pressure is considerably less than the critical pressure.

$$L = \sqrt{15A_0} \tag{131}$$

in which L = length of nozzle from throat to exit, in.
A_0 = area at the throat, sq. in.

The cross-sectional shape of nozzles may be either circular, elliptical, square, or rectangular with rounded corners. Experimental work indicates that the efficiency of a nozzle is practically unchanged by the shape of the cross-section when the following conditions exist: The internal surfaces must be smooth and the nozzle proportions made to

give the correct ratio of the area at the throat to that at the exit end. The length of the nozzle must be not less than, nor more than twice, the length as calculated by equation 131. Using a properly proportioned nozzle with straight sides throat to exit, the efficiency as measured by the velocity is about 96 to 97 per cent, corresponding to an equivalent energy efficiency of 92 to 94 per cent.

Example.—(*a*) Design a theoretically perfect nozzle to pass one pound of steam per second, the steam being initially dry and saturated at a pressure of 150 lb. per sq. in. abs. and the final pressure being 14.7 lb. per sq. in. abs. (*b*) If the turbine blades could be made to transform all of the issuing jet velocity into useful work, find the horsepower imparted to the rotor. (*c*) What is the theoretical steam consumption (water rate) in pounds per horsepower per hour?

Solution.—(*a*) From steam tables and Mollier (heat entropy) diagram, the following values are obtained:

$p_1 = 150$ lb. $\quad x_1 = 100$ per cent $\quad s_{g,1} = 1.569$

$p_2 = 14.7$ lb. $\quad x_2 = 87.02$ per cent $\quad s'_{g,2} = 1.569$

$p_0 = 87$ lb. $\quad x_0 = 96.15$ per cent $\quad s'_{g,0} = 1.569$

$v_{g,1} = 3.01$ cu. ft. $\quad h_{g,1} = 1193.5$ B.t.u. per lb.

$v'_{g,2} = 23.34$ cu. ft. $\quad h'_{g,2} = 1024.27$ B.t.u. per lb.

$v'_{g,0} = 4.86$ cu. ft. $\quad h'_{g,0} = 1149.47$ B.t.u. per lb.

From equation 129,

$$V_2 = 223.8\sqrt{1193.5 - 1024.27} = 2912 \text{ ft. per sec.}$$

and

$$V_0 = 223.8\sqrt{1193.5 - 1149.47} = 1485 \text{ ft. per sec.}$$

Using equation 130,

$$A_2 = \frac{1 \times 23.34}{2912} \times 144 = 1.154 \text{ sq. in.}$$

and

$$A_0 = \frac{1 \times 4.86}{1485} \times 144 = 0.471 \text{ sq. in.}$$

From equation 131,

$$L = \sqrt{15 \times 0.471} = 2.66 \text{ in.}$$

The entrance end of the nozzle will be rounded as shown in Fig. 197, the throat diameter is 0.77 in., the diameter of the nozzle exit is 1.21 in., and the length of the nozzle from throat to exit is 2.66 in.

(*b*) K.E. $= \dfrac{WV^2}{2g} = \dfrac{1 \times (2912)^2}{2 \times 32.17} = 131{,}800$ ft-lb. per sec.

Therefore

$$\text{Hp.} = \frac{131{,}800}{550} = 239.6.$$

(*c*) Steam per hp. per hr. $= \frac{3600}{239.6} = 15$ lb.

The exit end of a turbine nozzle is cut off obliquely, as shown in Figs. 195 and 198, in accordance with the angle made by the nozzle axis with the plane of blade rotation. This inclination is determined as a compromise between conflicting considerations. Higher theoretical efficiency can be obtained by using small nozzle entrance angles, as the blade action will be more efficient. Under these conditions a long nozzle is required and the frictional loss is greatly increased, the total deviation of the steam in the blading is increased, causing additional frictional losses, and the diagonally cut exit end of the nozzle is so wide that it fails to direct the steam into the blading properly. If this nozzle angle becomes too great, the blading loss becomes large, as the steam jet strikes the back of the blades at the entrance. Nozzle entrance angles used in practice range from 10 to 30 deg., 15 to 20 deg. being common.

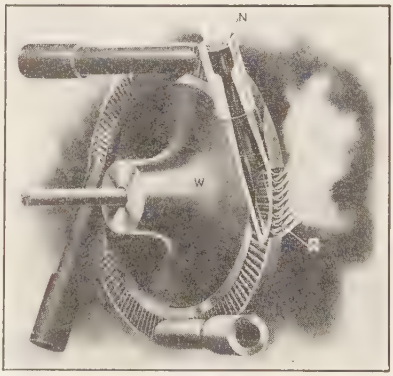


Fig. 198.—Principle of the impulse turbine.

239. Single-stage Impulse Turbine.—The simplest type of steam turbine consists principally of a single wheel revolving in a casing, the latter fitted with one or more nozzles. The De Laval " Class A " is the oldest turbine of this type; the essential elements of which are shown in Fig. 198. The steam is expanded from the initial to the exhaust pressure in nozzle *N*, and the kinetic energy of the jet is absorbed by a single row of blades *B* fastened to the periphery of wheel *W*. The blades deflect the jet as shown and absorb the greater part of the kinetic energy, so that the steam flows away from the wheel with a low velocity. Many turbines of this type are in use, but they are no longer manufactured because of the high rotative speeds, 10,000 to 30,000 r.p.m. For driving lower-speed machines it is necessary to use reduction gearing between the turbine shaft and the driven machine.

As many nozzles are used as may be required to make available the

amount of energy desired at full load, and provision is made for shutting off manually one or more nozzles as conditions may warrant. Governing for variations in load is effected by a throttling governor placed in the steam supply line near the turbine. By reducing the pressure of the steam admitted to the nozzles at light loads, the weight of steam discharged upon the blades is less than it is at a higher pressure, and correspondingly less energy is given to the turbine wheel to maintain a constant speed. The blades are made with bulb shanks, which are forced into slots milled in the circumference of the wheel. The outer ends of the blades have projections which fit closely together and are peened so as to form a continuous ring. Thus, if an individual blade becomes damaged, it may be removed and another put in its place. This type of turbine has been built in 15 sizes from 7 to 700 hp., with peripheral speeds as high as 1300 ft. per sec.

240. Action of Steam on Impulse Blades.—In an ideal impulse turbine the steam jet issuing from a nozzle would be brought to rest in the blades and would thus give up all its kinetic energy. In the actual turbine it is impossible to bring the jet to rest, owing to the limitations of practical designs. There is, therefore, always a loss in a turbine because of the residual or terminal velocity of the steam leaving the wheel.

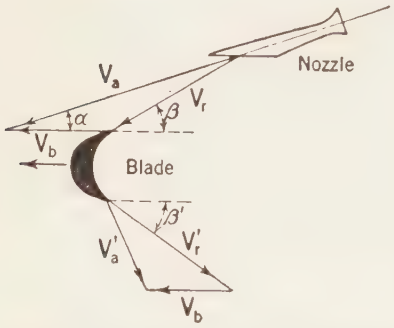


FIG. 199.—Velocity diagram for impulse turbine blading.

Figure 199 shows a section of a blade in a single-stage impulse turbine. If the minimum eddy loss at entrance is to be obtained at the point at which the steam jet enters the blade, the jet must enter along a tangent to the inside curve at the entrance edge. This direction is shown by the line V_r in the figure, and if the blade were stationary, the steam jet would move as shown by V_r. Since the blade moves ahead and away from the jet, the steam must be directed as indicated by V_a in order to strike the blade in the direction indicated by V_r.

In the case of steam impinging on a blade as shown in Fig. 199, the nozzle directs the jet against the blade with absolute velocity V_a. A nozzle angle α (about 20 deg. with the plane of the wheel) is used. Since the blade moves at a velocity of V_b, the velocity V_r of the steam jet relative to the blade is the geometrical resultant of V_a and V_b. For convenience in construction, the entrance angle β and exit angle β' are made equal. Neglecting frictional losses in the blade, the relative exit velocity $V'_r = V_r$ and the resultant of V'_r and V_b is the absolute exit

velocity V'_a. Obviously, the energy removed from steam passing over the blade is

$$\frac{WV_a^2}{2g} - \frac{W(V'_a)^2}{2g}.$$

If the blade were theoretically perfect the absolute velocity V'_a would be zero, and all the kinetic energy would then be removed. This is practically impossible in an actual turbine, and there is always a loss due to the residual velocity V'_a. The best that can be done is to choose the nozzle angle α and the velocity of the blade with respect to the steam jet velocity so that the numerical value of V'_a will be a minimum. *For the production of maximum work the turbine blade velocity should be about one-half of the absolute velocity of the steam jet.*

241. Steam-turbine Staging.—When the blades of a turbine are operated at peripheral speeds equal to half the steam jet velocities, one of two difficulties arises. The stresses induced in the rotor may become excessive or the rotative speed may be too high for direct connection to the machine which is to be driven. The turbine rotor may be operated at or near the theoretically desirable speed and the power transmitted to the driven machine through gears which decrease the turbine speed to the necessary extent. However, the gears necessary for the desired speed reduction or the size of the turbine wheels would be excessive in large units. In order to decrease the peripheral velocity of the turbine rotor, and at the same time use more of the energy of the steam, more than one set of rotor blades is employed. This is called compounding or staging; two methods are in use, *velocity staging* and *pressure staging.*

A turbine employing both principles is illustrated by Fig. 200. The velocity developed in the first-stage nozzles C is utilized in two rows of moving blades D and F. Blades D absorb about one-half of the velocity, the steam escaping from them is reversed by stationary blades E, and directed into the second row of moving blades F, which absorb the remainder of the first-stage velocity. The steam is then expanded in the entrance nozzles of five successive pressure stages, each of which has one row of moving blades, i.e., one velocity stage per pressure stage. Each set of nozzles utilizes a portion of the available heat, which when transformed into velocity results in a steam-jet speed of about twice the rotor speed, so that the kinetic energy is absorbed by the single row of blades.

The multivelocity absorption of energy of the nozzle discharge, as shown in the first stage of Fig. 200, is called *velocity staging* and is commonly known as the Curtis principle. Steam partially expanded in a set of nozzles, and the resulting velocity utilized in one row of

moving blades, may be expanded further in a second set of nozzles with an increase of steam velocity. Moving blades adjacent to the second set of nozzles absorb the velocity, as in the first set. This expansion and increase of velocity may be repeated several times. The process is termed *pressure staging* and is generally called the Rateau principle. Figure 200 shows the method for six sets of nozzles in six pressure stages, the first of which has two velocity stages, the other five have one velocity stage each. A complete turbine may be composed of any combination of Curtis stages and Rateau stages.

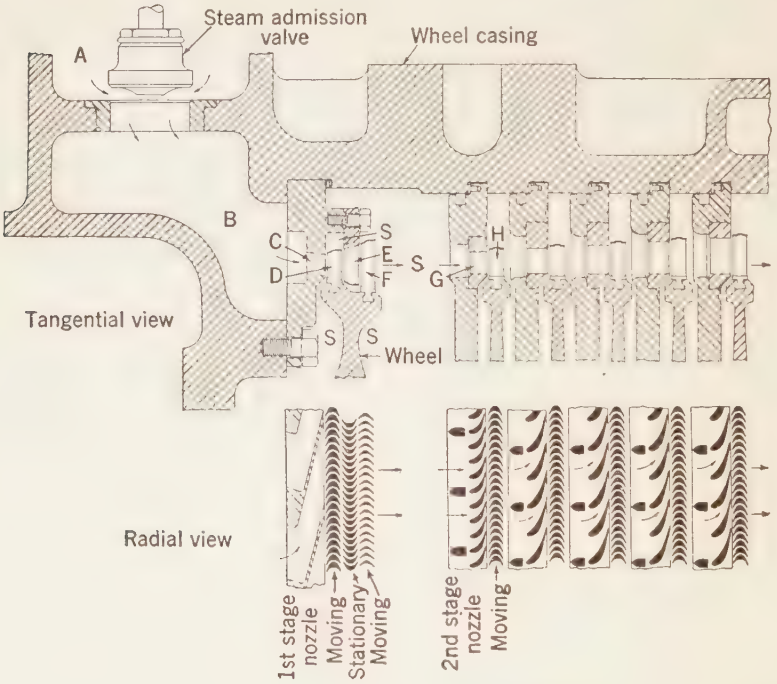


FIG. 200.—Compound impulse turbine with six pressure stages; the first of which has two velocity stages, the other five have one velocity stage each.

The energy-transforming capacity of the Curtis stage is greater than that of the Rateau, with less stages and cheaper construction, but the Rateau principle is more efficient in the transformation. All impulse turbines designed for high efficiency use either Rateau staging following a Curtis stage, or Rateau staging throughout. The turbine shown in Fig. 200 offers the advantages of a large pressure drop in the first-stage nozzles C whereby the steam enters the casing of the turbine at a lower pressure than if the first stage had been of the Rateau type. In this way the problem of design of the shell and separating diaphragms

(nozzles) is simplified. The number of stages that a multipressure turbine should contain must be such that the energy-absorbing capacity of the rotor is equal to the available energy of the steam. The energy-absorbing capacity is determined by the diameter of the rotor, revolutions per minute, kind of staging, and number of stages.

Example.—A steam turbine of the impulse type is to be used for driving a generator at 3600 r.p.m.; the mean diameter of the wheels is to be 48 in. The quantity of dry steam to be supplied is 10,800 lb. per hour at a pressure of 200 lb. per sq. in. abs. Assuming adiabatic expansion to a condenser pressure of 2 in. of mercury abs., and neglecting friction, find (*a*) the theoretical number of velocity stages required, (*b*) the theoretical number of pressure stages, and (*c*) the number of pressure stages if there are two velocity stages for the first pressure stage.

Solution.—(*a*) *Velocity staging.*

$$\text{Peripheral velocity} = \frac{3600}{60} \times \frac{48}{12}\pi = 754 \text{ ft. per sec.}$$

$$\frac{10{,}800}{60 \times 60} \times (1197.8 - 861.7) = 1008.3 \text{ B.t.u. per sec. available.}$$

$$\text{K.E.} = \frac{WV^2}{2g},$$

or

$$1008.3 \times 778 = \frac{3V^2}{64.34},$$

thus

$$V = 4102 \text{ ft. per sec. available.}$$

Steam velocity per stage = $2 \times 754 = 1508$ ft. per sec., thus

$$N_v = \frac{4102}{1508} = 2.72.$$

Use 2 velocity stages.

(*b*) *Pressure staging.* Steam velocity to be absorbed per stage = 1508 ft. per sec.

$$\text{K.E.} = \frac{WV^2}{2g} = \frac{3 \times (1508)^2}{64.34}.$$

$$\text{K.E.} = 106{,}033 \div 778 = 136.3 \text{ B.t.u. per stage.}$$

Heat drop = 1008.3 B.t.u. available; thus

$$N_p = \frac{1008.3}{136.3} = 7.4.$$

Use 7 pressure stages.

(*c*) *Curtis stage (double wheel) and Rateau staging.* Steam velocity required for double Curtis stage $= 2 \times 2 \times 754 = 3016$ ft. per sec.

$$\text{K.E.} = \frac{WV^2}{2g} = \frac{3 \times (3016)^2}{64.34};$$

K.E. $= 424{,}134 \div 778 = 545.2$ B.t.u. for Curtis rotor.

Heat available for Rateau stages $= 1008.3 - 545.2 = 463.1$ B.t.u.

From part *b*, each Rateau stage absorbs 136.3 B.t.u., thus

$$N_r = \frac{463.1}{136.3} = 3.4 \text{ Rateau stages.}$$

The turbine would consist of 1 Curtis and 3 Rateau stages.

In making a comparison of the results obtained in the preceding example, it might seem that velocity staging is the better, because a smaller number of stages is required. As stated previously in this article, the advantage of the smaller number of stages is overbalanced by the fact that pressure staging is more efficient. Turbines with pressure staging are used extensively, and are often called multicellular because each pressure stage is composed of a compartment that is steam tight except for the openings through the nozzles. Where no difference in pressure exists, as in velocity staging, there is no leakage. In the velocity-stage type, the pressure beyond the nozzles is the same throughout the entire turbine casing and thus there is less interstage leakage. Curtis rotors of three velocity stages may be used, but generally there are not more than two because the high-velocity steam jet is broken up by its passage through the blades and the efficiency is low after the first two stages. A very high entering steam velocity also produces "skin" friction between the steam and the blade surfaces, causing a considerable loss.

242. Single-pressure, Compound-velocity-stage Impulse Turbines.—The Elliott, De Laval "Class C," Moore single-stage, General Electric single-stage, and Terry non-condensing are well-known steam turbines of this class. Figure 201 shows a longitudinal section through an Elliott single-pressure, two-velocity-stage, non-condensing turbine embodying the "Curtis" (velocity staging) principle. The rotor consists of a shaft, two wheels, and the speed governor. The revolving blades and stationary guide vanes are made of nickel steel and are inserted in dovetail slots in the wheels and stationary holder. The nozzles are of the diverging type and are machined in a nozzle ring casting bolted to the side of the steam ring. The spacing is very close so that the steam jets unite upon leaving the nozzles, and the blades, therefore, receive a continuous band of steam. The intermediate

stationary guide vanes receive the steam from the blades of the first wheel and direct the steam upon the second row of blades. As complete expansion takes place in the nozzles, uniform pressure exists throughout the remainder of the turbine; hence there is no tendency for leakage of steam over the tops of the blades. This fact permits the use of large running clearances. Steam leakage along the shaft is partially prevented by carbon packing rings, Fig. 201.

The speed of the Elliott turbine is controlled by a centrifugal governor mounted on the end (left end in Fig. 201) of the main shaft and attached, through suitable linkage, to a throttle valve in the steam supply line. Two governor weights are mounted upon hardened-steel

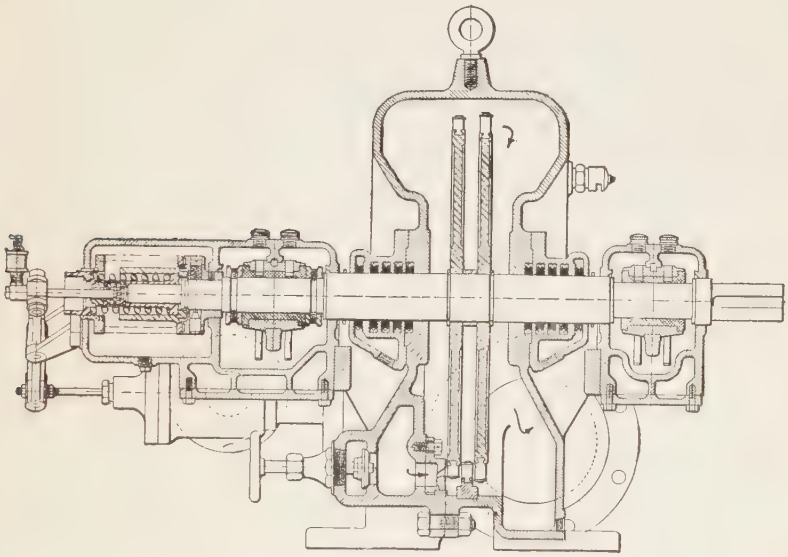


Fig. 201.—Longitudinal section, two-velocity-stage steam turbine.

knife-edges in a bronze cage that rotates, the cage itself being keyed and pressed on an extension of the turbine shaft. A coil spring resting against a sliding sleeve loads the governor so that any change in the spring compression varies the speed of the turbine. The motion from the weights to the governor linkage is transmitted by a hollow spindle that does not rotate; this permits positive lubrication for the governor mechanism.

Machines suitable for high speeds, such as small high-head centrifugal pumps, centrifugal blowers and compressors, and electric generators, can be directly connected through a flexible coupling to steam turbines of this class. Reducing gears are used between the turbine and

machines restricted to speeds lower than those of the turbine, as medium-size generators, large centrifugal pumps and blowers, belt pulleys, and paper mill and other moderate-speed machinery. Elliott turbines of the type shown by Fig. 201 are available in sizes up to 150 b.hp.

The De Laval " Class C " turbine differs essentially from the Elliott turbine in that the rotor consists of a single wheel fitted at the periphery with two or three rows of blades. The De Laval turbine of this class is built in sizes up to 800 b.hp. and is primarily designed for non-condensing service.

The Terry non-condensing turbine applies the principles of single-pressure, compound-velocity-stage turbines to a single wheel of the re-entrant (helical flow) type. The steam issues from an expanding nozzle at high velocity, enters the side of the wheel bucket and impinges against a blade, where the direction of steam flow is reversed 180 deg., Fig. 202. As this single reversal uses but a portion of the available energy, the steam is received in a reversing chamber and returned to the wheel. The reversal of the steam several times to the same row of moving buckets gives the desired multivelocity effect, and the speed of rotation is reduced. The moving buckets are semicircular and are milled in the solid rim of the wheel. The stationary reversing buckets are arranged in groups around the inner surface of the turbine casing, each group being supplied with a separate nozzle, Fig. 203. The nozzles have hand-controlled valves. These permit the use of the required number for high efficiency at various loads. Each stationary bucket has a crescent-shaped hole at the center, Fig. 203, through which the steam enters the casing exhaust chamber, when the steam velocity has been sufficiently reduced. The casing is subject only to the exhaust pressure and temperature and is in reality a portion of the exhaust piping. The governor is of the throttling type, similar to the governor shown in Fig. 201.

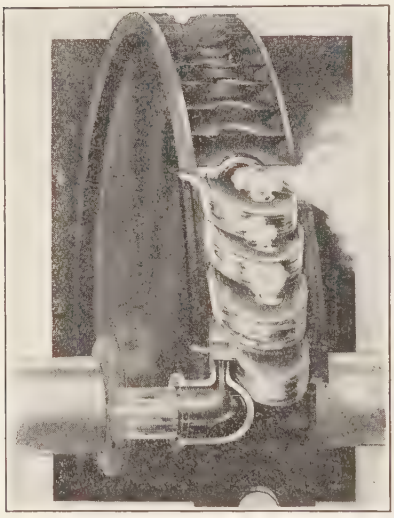


Fig. 202.—Action of steam in Terry re-entrant type impulse turbine.

As Terry turbines of this type are built only in sizes from 5 to 1000 hp., their use in central stations is practically confined to driving auxiliary apparatus, such as pumps, blowers, and small generators.

The non-condensing turbine is used when exhaust steam can be utilized for heating purposes or for process work in sugar plants, paper mills, tanneries, rubber mills, etc. The absence of oil in the exhaust steam makes the non-condensing turbine particularly valuable in such plants. The turbine acts as a power-generating reducing valve and abstracts only a small percentage (about 5 per cent) of the heat in the steam.

243. Compound-pressure, Single-velocity-stage Impulse Turbines.—Professor Rateau, of Paris, was the pioneer in the development of this type of steam turbine. The turbine shown by Fig. 204 is illustrative of the Rateau type of construction. The rotor consists of a number of

FIG. 203.—Phantom view of Terry re-entrant type impulse turbine.

steel disks, keyed and shrunk on the turbine shaft and held in place axially by shrink rings which fit into recesses on both sides of each wheel. The blades are rigidly fastened to the periphery of the wheel by bulb shanks, individually driven into slots and peened into place. The stationary element consists of the steam heads, diaphragms, nozzles, etc., as shown in Fig. 204. The cylindrical sections, end castings, and diaphragms are all split on the horizontal center line so that the entire upper half of the casing may be removed for inspection without disturbing the steam and exhaust connections. Steam or air leakage along the shaft is prevented by three-part arch-bound carbon rings held

in place by non-corrosive metal springs and prevented from rotating by stops. The circular diaphragms contain the nozzles and separate the stages.

Steam admitted to the steam chest of the turbine passes through the first-stage nozzles; they occupy only a part of the circumference and may be controlled by hand-operated valves. The kinetic energy of the steam is absorbed by the blades on the first turbine wheel. After leaving the first row of moving blades at a low velocity, the steam is again partially expanded in the second set of nozzles and velocity is acquired. The second wheel absorbs this velocity and the steam passes on through the following stages, each consisting of a successive row of nozzles and blades; ten pressure stages are shown in Fig. 204. At the high-pressure end of the turbine there are only a few groups of

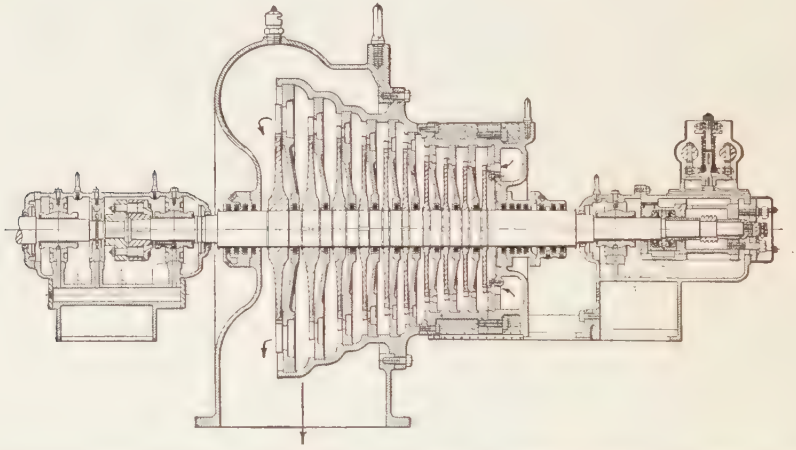


Fig. 204.—Longitudinal section, impulse turbine with pressure staging; Rateau type.

nozzles, but in each successive stage the number increases to allow for the increased volume of the expanded steam, until in the last stages the entire circumference of the diaphragm is used. The height of the blades increases as the exhaust end is approached, because of the increase in the volume of steam flowing.

Under ordinary conditions of operation, no appreciable end thrust exists with turbines of this type. However, the rotor must necessarily be located in a predetermined position with respect to the stationary elements. This is accomplished by using a Kingsbury thrust bearing, right end, Fig. 204, made up of a number of tilting, self-aligning bronze shoes which bear against a collar keyed in a fixed position on the turbine shaft. As the collar rotates, the shoes tilt and allow a thin film of oil to be drawn in between the shoes and the collar. The load is

thus "floated" on a film of oil separating the bearing surfaces. Two sets of shoes are provided, one set on each side of the thrust collar to take thrust in either direction. The bearing housing rests upon a sliding fit, which permits endwise adjustment of the rotor while the turbine is in operation.

The governor mechanism is of the oil-operated type. The main governor, right end, Fig. 204, is of the spring-loaded centrifugal type with spherical weights and is gear-driven from the turbine shaft. The governor motion is transmitted through linkage to the pilot valve of an oil-relay cylinder, which admits oil under pressure to either side of the main operating piston. The movement of this piston opens or closes

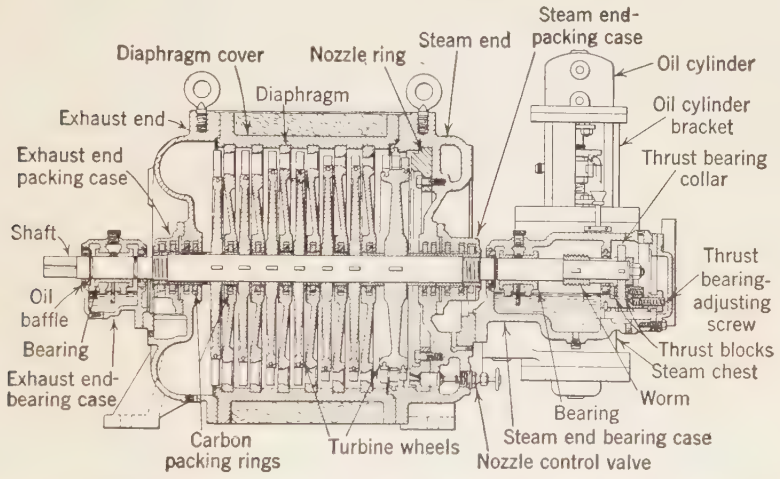


FIG. 205.—Compound impulse turbine with seven pressure stages; the first of which has two velocity stages, the other six have one velocity stage each.

a double-seated balanced governor (throttle) valve which controls the admission of steam to the turbine, and hence the speed.

244. Compound-pressure, Compound-velocity-stage Impulse Turbines.—The advantages of velocity staging are: (1) a large pressure drop can be employed in each stage for a given blade velocity, (2) there is a small number of stages for a given heat drop and consequent reduction in first cost, and (3) the maximum pressure and temperature in the turbine casing are less than in pressure staging, thus decreasing stresses and interstage leakage. A high steam velocity as used in multistage velocity turbines is, however, a disadvantage, especially if moisture is present in the steam, as rapid erosion of the blades is inevitable. In multistage pressure turbines the stage velocity is constant but the steam pressure is gradually decreased from inlet to exhaust.

The efficiency of a purely velocity-compounded turbine is less than that of a purely pressure-compounded turbine, but the larger pressure drops permissible with the former type are an advantage. In order to utilize the low spouting velocity of the steam and higher efficiency of the pressure type, and also reduce the number of stages, both pressure and velocity staging are used in the same turbine. Ordinarily there are two velocity stages in the first pressure stage and one velocity stage for each of the remaining pressure stages.

An example of this type of turbine, as made by the Moore Steam Turbine Corporation, is shown in section by Fig. 205. . The steam is expanded through the first-stage nozzles, and impinges on a Curtis

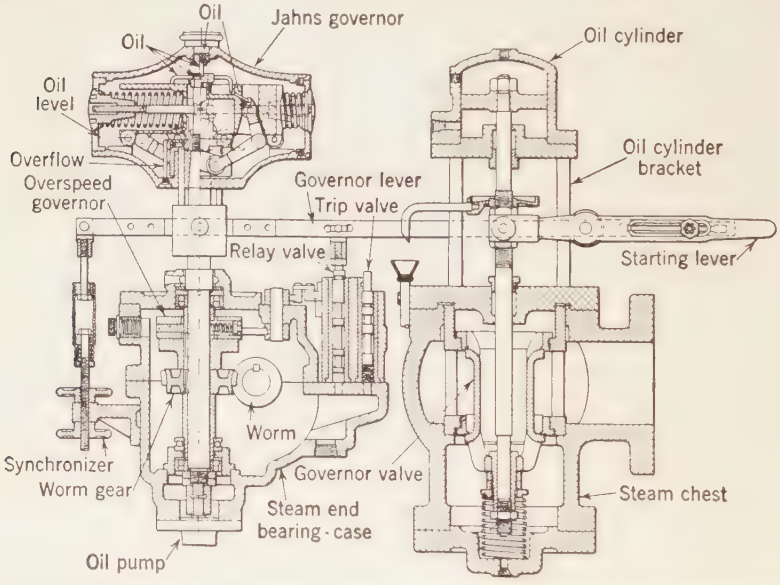


Fig. 206.—Centrifugal oil-pressure relay-type throttling governor and valve gear.

(velocity-stage) wheel consisting of two rows of moving blades with a stationary row of reversing guide vanes between them. After this first-pressure stage, the steam flows successively through a series of six additional pressure stages, each consisting of a set of nozzles and a turbine wheel. As the steam passes through successive stages the passage areas increase sufficiently to provide for the increased volume of steam.

The casing of the Moore turbine is split horizontally so that the upper half can be removed to give access to the interior, and is encased with a layer of non-conducting material covered by sheet metal. The wheels are keyed and shrunk on the shaft, which is supported by two ring-oiled bearings. Carbon packing is used for sealing against inter-

stage and other leakage. A Jahns centrifugal governor, Fig. 206, mounted on a vertical shaft, is driven by a worm gear from the turbine shaft (right end in Fig. 205). The governor actuates the oil-relay valve which admits oil under pressure above or below the piston in the oil cylinder. This piston in turn controls the opening of the throttling governor valve. Oil under a pressure of 35 to 40 lb. per sq. in. is supplied by a geared type of pump driven from the lower end of the governor spindle. Hand nozzle-control valves are furnished: (1) for overload operation, (2) for full-load operation on steam pressures lower than the pressure for which the turbine was designed, and (3) economical operation at fractional loads.

245. Velocity Diagram for Compound-velocity-stage Impulse Turbine. No Losses.—The extended velocity diagram for a single-pressure, single-velocity-stage steam turbine is shown by Fig. 199. The velocity diagram for a single-pressure two-velocity-stage turbine is shown by Fig. 207. The blade velocity being the same for each turbine wheel, the series of velocity triangles have been superimposed to form one diagram. CA, the base of the velocity triangles, represents the blade velocity V_b in feet per second, while AB is the initial steam velocity, the nozzle angle being α. CB is, therefore, the relative inlet velocity, and the angle β (BCM) is the inlet angle of the first row of moving blades. Since there are no losses, the relative exit velocity CD is equal to CB; and since the blades are equiangular, the angle α' is made equal to angle β. This is easily accomplished by making the axial components of the velocities equal. DA is the absolute velocity of the steam leaving the first row of moving blades, or the velocity relative to the stationary guide vanes. Hence, angle β' (DAN) is the inlet angle of the guide vanes. Assuming no losses through the guide vanes, the absolute entrance velocity EA is equal to DA, and angle EAM is the outlet angle of the guide vanes. The velocity triangles for the second row of moving blades may be drawn in the same manner as for the first row. For the most efficient performance of work the final absolute vector AF should be perpendicular to the direction of motion of the blade, that is, AF and AH should be coincident. V_2 represents the tangential change of velocity or the "velocity of whirl" of the steam in the second row of moving blades. Any number of pressure stages may be analyzed in a similar manner.

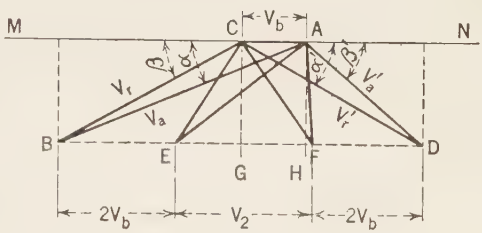


Fig. 207.—Velocity diagram for two-row velocity-stage impulse turbine.

246. Reaction Turbines.—In the pure reaction turbine the expansion of the steam occurs in the moving blades. Turbines commonly called reaction turbines use both the impulse and reaction principles. This type of reaction turbine is shown diagrammatically by Fig. 196. The steam enters from the left and passes through a set of stationary blades (stator) in which occurs some drop in pressure. In passing through the next set of blades, which are moving, a further drop in pressure occurs. The velocity acquired in the stationary blades and also that produced in the rotor blades are absorbed by the rotor blades. The jet with low residual velocity is deflected to the second set of stationary blades with another drop in pressure (the reduction seldom exceeds 3 lb.), and then enters the second rotor at practically the same velocity as that of the blades. Considerable expansion takes place in the moving blades, and consequently, because the pressure is not the same on both sides of these blades, the turbine is called a *reaction turbine.*

Example.—A steam turbine of the reaction type is to be used for driving a generator at 3600 r.p.m.; the mean diameter of the rotor is to be 48 in. Assuming adiabatic expansion and neglecting friction, find the number of stages required in a "pure" reaction turbine and in an "impure" reaction turbine if 10,800 lb. of dry steam are supplied per hour at a pressure of 200 lb. per sq. in. abs., the condenser pressure to be 2 in. mercury abs.

Solution.—*In order to obtain the highest efficiency, the velocity of the moving blades of a reaction turbine should be approximately the same as that of the steam relative to the blades.*

Blade velocity, $V_b = \frac{3600}{60} \times \frac{48}{12} \pi = 754$ ft. per sec., therefore the steam velocity = 754 ft. per sec.

$$\text{K.E.} = \frac{WV^2}{2g} = \frac{3 \times (754)^2}{2 \times 32.17} \times \frac{1}{778};$$

K.E. = 34.1 B.t.u. heat drop per stage per sec.

$$\text{Total heat drop} = (1197.8 - 861.7) \times \frac{10{,}800}{60 \times 60} = 1008.3 \text{ B.t.u. per sec.}$$

$$\textit{Pure} \text{ reaction stages} = \frac{1008.3}{34.1} = 29.$$

For the pressure-stage impulse type (with the same data) in the illustrative problem of Art. 241, the number of stages required was 7. Since the "impure" or actual reaction turbine employs a combination of the impulse pressure-stage and the reaction principle, the number of stages may be taken as a mean, or

$$\textit{Impure} \text{ reaction stages} = \frac{29 + 7}{2} = 18.$$

The preceding example and solution show that the reaction turbine requires many more stages than the impulse type. This is due to the fact that a much higher ratio of blade velocity to steam velocity is required, for the best efficiency of a reaction stage, than is required for an impulse stage. However, the loss due to friction is less in the reaction type since the steam velocities are comparatively low; also erosion of the blading is reduced to a minimum. The axial-flow impulse reaction turbine, Fig. 208, is often called the Parsons turbine, after its inventor, Sir Charles A. Parsons. The Allis-Chalmers Manufacturing Company and the Westinghouse Electric and Manufacturing Company are the important American manufacturers of this type of turbine. In the radial-flow reaction turbine, the steam flows outward from the shaft to the periphery of the rotor, in this way accommodating the increased volume due to expansion. The Ljungström (Swedish) is the best known of the radial-flow type of reaction turbine.

The 15,000-kw. Westinghouse tandem-compound turbine shown by Fig. 208 operates at 3600 r.p.m. with steam conditions at 450 lb. per sq. in. and 825 deg. F. This turbine consists of a *single-flow* impulse and reaction high-pressure cylinder, and a *double-flow* reaction low-pressure cylinder. Each unit consists essentially of a fixed casing or cylinder and a revolving (enlarged) spindle or drum. The ends of the spindle are extended in the form of a shaft and carried in suitable bearings. Steam enters the turbine at A after passing through the throttle valve, which is under the control of the governor. Steam passes through the nozzles to a two-moving-row velocity-compound impulse stage and thence through three groups of reaction blading to the high-pressure turbine exhaust at B. The steam is then led through an overhead receiver pipe to C where it enters the low-pressure cylinder which employs only reaction staging.

Each row of reaction blades, both stationary and revolving, extends completely around the turbine, and the steam flows through the full annulus between the spindle and the cylinder. In the low-pressure cylinder the length of the blades increases continuously and gradually from the steam inlet C to the exhaust D. The same effect is produced in the high-pressure cylinder by making the rotor in steps; there are generally three such reaction steps, as shown in Fig. 208. The reaction blades in each step are arranged in groups of increasing length. At the beginning of each of the larger steps the blades are usually shorter than at the end of the preceding smaller step, the change being made so that the correct relation of blade length to spindle diameter is secured.

The difference in steam pressure at the end of each step produces a thrust tending to force the spindle toward the left, Fig. 208. This thrust is counteracted by two Kingsbury thrust bearings and the bal-

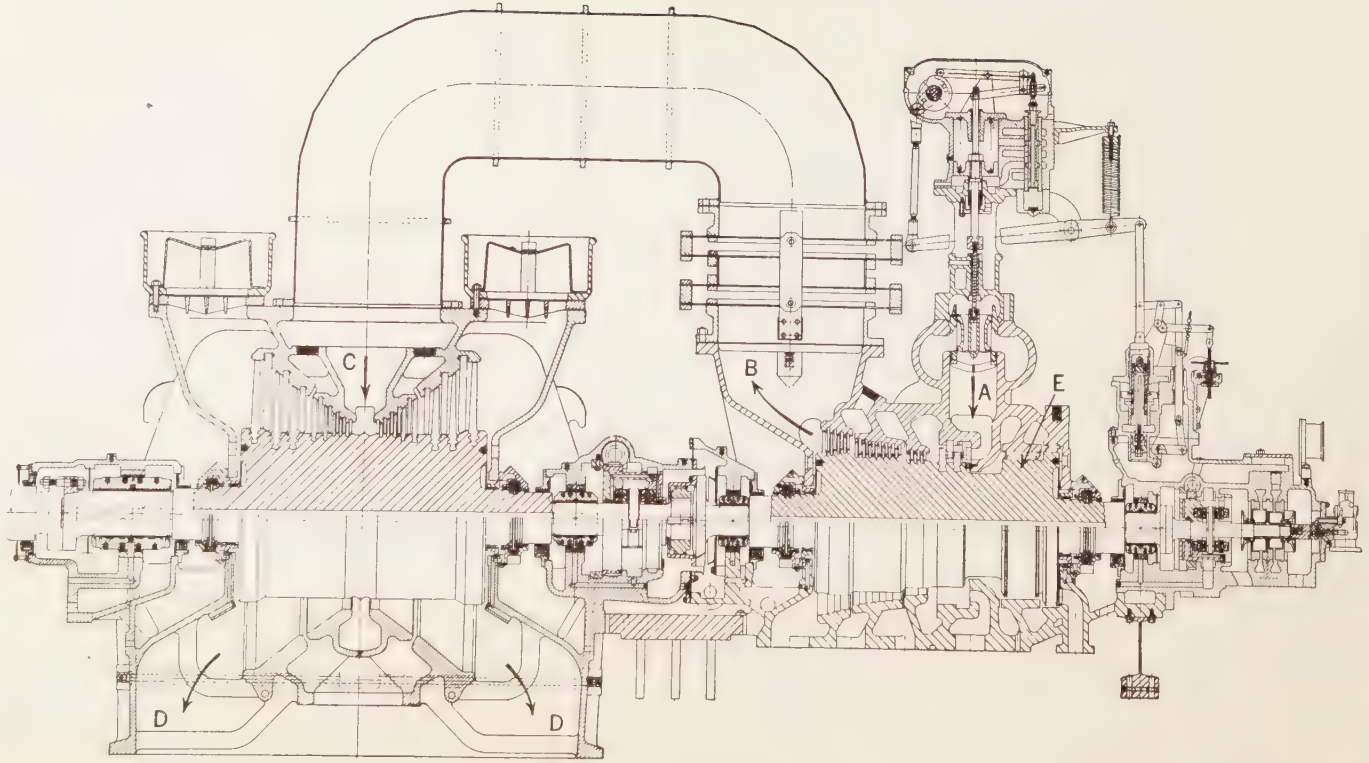


Fig. 208.—Westinghouse tandem-compound steam turbine; high-pressure cylinder (right) has two velocity impulse stages and three groups of reaction blading, double-flow low-pressure (left) cylinder has reaction stages only.

ancing piston E. This enlargement of the spindle is called a piston, although it does not come in contact with the cylinder, but both the piston and the cylinder are provided with alternate rings which form a labyrinth packing to retard the leakage of steam. Equalizing passages are provided by which steam of appropriate pressure may act on the piston to partially counterbalance the end thrust to the left. Both turbine thrust bearings are provided with means for adjustment of the axial blade-clearances.

The *velocity diagram* for the first stage of a reaction turbine is shown by Fig. 209. In a reaction stage there is an increase of relative velocity between entrance and exit, due to the pressure drop in the blade channels. The fixed and moving rows have identical blades, and the heat drop for the stage is divided equally between the stationary and moving rows of blades. Assuming the same friction losses in identical rows, it follows that, with the same exit angle for each, $\alpha = \alpha'$, the entrance and exit triangles are identical, as shown in Fig. 209. Thus $V_r = V'_a$ and $V_a = V'_r$. From the velocity diagram it is evident that the axial

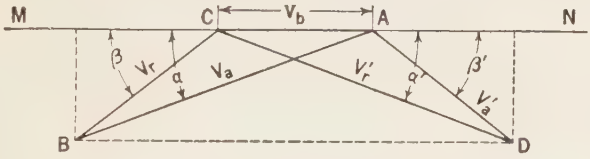


Fig. 209.—Velocity diagram for reaction blading.

components of the steam velocity are equal and the dynamical thrust is zero. With the usual construction, however, in which the blade heights are kept constant throughout each step or group, there is a progressive increase of velocity and consequently a slight thrust. Normally, this thrust is small and is neglected. There is a statical thrust, which does not occur in the impulse turbine, due to the difference of pressure between entrance and exit, at each rotor. The cumulative effect of these statical thrusts is counteracted by balance pistons, as shown at E in Fig. 208.

247. Combination Steam Turbines.—The high-pressure portion of the reaction turbine is comparatively inefficient because of blade tip leakage; the low-pressure part is usually efficient. The impulse turbine has approximately the same efficiency at all loads, if governing by nozzle control is used. The replacement of the high-pressure section of the reaction turbine by impulse staging very materially shortens the rotor, thus increasing its stiffness. Distortion of the turbine casing and rotor is avoided by such an arrangement, as the entering steam is considerably reduced in temperature and pressure in the nozzles before

reaching the rotor. A combination turbine is one in which impulse blading is used for the high-pressure stages and reaction blading in the low-pressure stages.

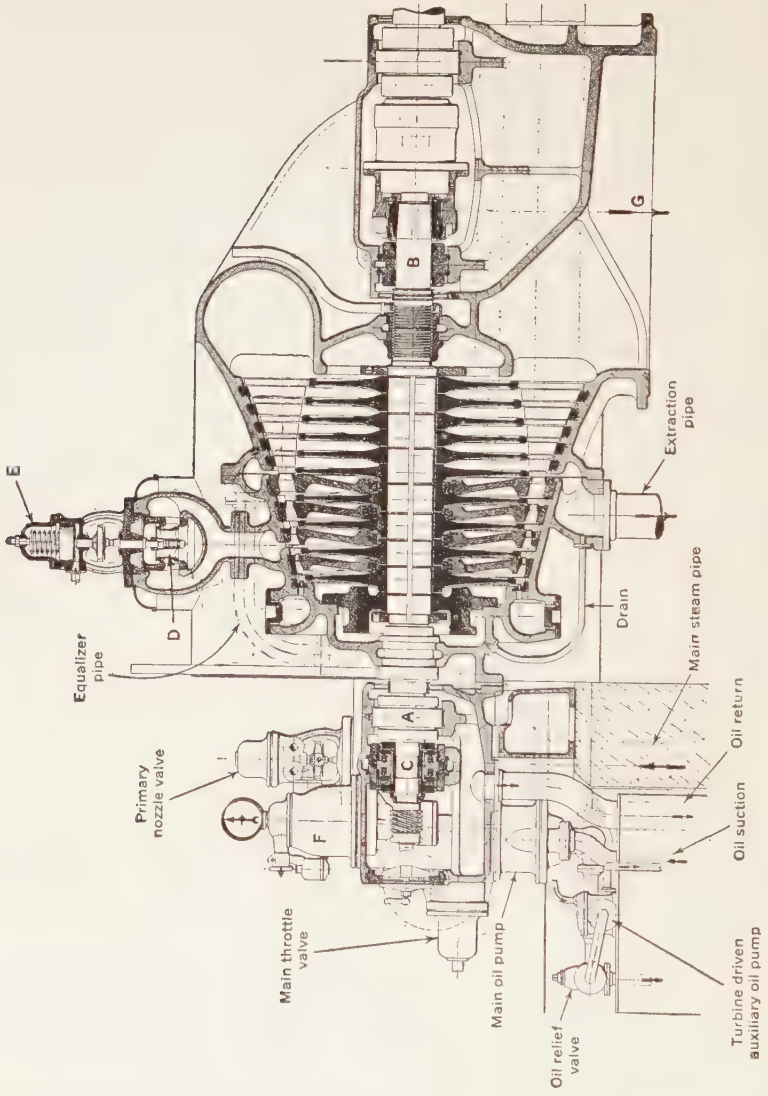


FIG. 210.—Longitudinal section, 20,000-kw. combination impulse and reaction turbine.

Figure 210 shows a section through a combination turbine, which is typical of some recent installations of large single-cylinder machines. Steam passes first through five single-row impulse wheels separated by diaphragms in the usual manner, and then through five reaction wheels,

each carrying one row of blades. In the smaller combination turbines, 1000 to 10,000 kw., a high-pressure impulse wheel of comparatively large diameter may be used, with one or more velocity stages and partial peripheral admission. The reaction section which follows has full admission, and the diameter of the drum is comparatively small. The drum type rotor, similar to the rotor of Fig. 208, is employed in these (smaller) turbines. A solid-type rotor, such as is used in turbines for small outputs, would not be suitable for large turbines, as the stresses would be high because of the increased diameter of the rotor. To prevent excessive rotor stresses and to realize the efficiency of the conical-flow-type turbine, the reaction blading is placed on disks mounted on the turbine shaft, Fig. 210. In the larger turbines the volume of steam in the reaction stages is very great so that the diameter at the low-pressure end of the turbine must be enlarged in order to insure sufficient sectional area for the steam passage. The dimensions of the large combination turbines are comparatively small, particularly the rotor, the effective length of which is 42 in. for a 15,000-kw. unit running at 3600 r.p.m. The short length is mainly attributable to the absence of diaphragms in the reaction staging, so that the distance between the individual disks is made only large enough to be consistent with strength.

Large turbines require special precautions to prevent unequal expansion of the casing. The turbine illustrated by Fig. 210 is provided with a number of ribs arranged to counteract any tendency to distortion due to unequal temperatures at the inlet and exhaust ends of the casing. The main bearings *A* and *B* (babbitt lined) are supplied with oil under a pressure of 40 lb. per sq. in. from an oil pump bolted to the bottom of the turbine casing and driven by a worm gear on the left end of the turbine shaft, Fig. 210. A segmental thrust bearing *C*, to take up any uncompensated axial thrust, is also carried by the bearing pedestal at the high-pressure end of the casing. Only one balance piston, placed left of the first wheel, Fig. 210, is necessary on account of the location of impulse stages in the high-pressure part of the turbine. Labyrinth shaft packing glands are provided at the admission and exhaust ends of the turbine. Both glands are normally under vacuum and are continuously supplied with a small amount of live steam in order to prevent air from entering the turbine. The blades of the impulse part of the turbine are made of nickel steel, and, being subjected to superheated steam, they corrode very little. The moving blades of the low-pressure reaction stages are made of stainless steel (14 per cent chromium), and the stationary reaction blades are made of brass.

The turbine, Fig. 210, is equipped with two main automatic throttle valves. These valves are so arranged that they are closed simultaneously

by an automatic tripping device, in case of overspeed. Steam is supplied through automatically operated double-beat primary nozzle valves (usually three), actuated by the piston in an oil-relay cylinder, similar to *E*. The steam next passes through the nozzles to the impulse blading and is then expanded in the reaction sections to the condenser pressure and exhausted to the condenser through *G*. Overload valve *D* is provided for admitting steam between the second and third impulse stages. Valve *D* opens when the primary nozzle-valves cannot pass a sufficient quantity of steam to carry momentary overloads, to maintain a constant speed, and to enable the turbine to carry the load should the steam pressure drop below normal. The turbine is equipped with an extraction pipe for the removal of some of the expanded steam at about atmospheric pressure, and the extracted steam is used for heating or other industrial purposes (Art. 250). The enclosed centrifugal governor *F* controls the two main throttle valves, the primary nozzle valves, and the overload valve *D*. Governor *F* is driven from the main turbine shaft by worm gearing. Oil under pressure from the lubricating oil-pump fills the pipe leading to the space beneath the piston at *E*. The pipe on the supply side is connected to a chamber formed by an adjustable sleeve on the governor spindle. The sleeve can cover and uncover an oil by-pass port, according to the speed fluctuation, and the governor sleeve can be raised or lowered in consequence. A quantity of oil escapes, and the oil pressure under the piston *E* is changed accordingly. When the sleeve on the governor spindle reduces the escape area of the by-pass port the oil pressure under the piston increases. The increased pressure forces up one or more of the relay pistons, according to the predetermined setting of the control springs, opens the nozzle valves and overload valve successively, and admits the necessary steam to meet the increased load on the turbine. One notable feature about the relay governing system on this turbine is the absence of lever connections between the governor *F* and the oil-relay cylinders of the valves. The equalizer pipe allows the equalization of the steam pressure on the outboard side of the dummy piston with the pressure in the intermediate portion of the turbine. Any steam leakage past the dummy piston flows through the equalizer pipe to the intermediate portion for further utilization in the later stages of the turbine.

248. Multicylinder Steam Turbines.—Multicylinder or compound turbine is the term applied to turbines in which the steam passes in sequence through one or more separate units (cylinders), partially expanding in each. The steam passes first through the high-pressure cylinder and then is led to the low-pressure cylinder. If the high-pressure and low-pressure units are placed on the same shaft, the machine is called a *tandem-compound* turbine. Steam turbines are

being built with three units in tandem. Sometimes better results are obtained by mounting each unit on its own shaft and running the two at different speeds. With the latter arrangement, the machine is called a *cross-compound* turbine. In the compound design the temperature range is less in each cylinder than in a single-cylinder turbine operating between the same initial and final steam conditions, and the stresses in the casing and rotor are decreased accordingly. Three units in series may also be used, the high-pressure and intermediate-pressure cylinders usually being in tandem and the low-pressure unit in a separate cylinder driving a separate generator. Multicylinder turbines are also being built with a *vertical-compound* arrangement in which the high-pressure turbine and its generator are mounted on top of the low-pressure turbine and its generator.

Dry steam is desired throughout the range of expansion in any turbine. The use of multicylinder machines may necessitate reheating of the steam to increase its temperature after passage through a part of the turbine. A 50,000-kw. Westinghouse turbine, at the Crawford Avenue (Chicago) Station, has the high-pressure and intermediate-pressure cylinders arranged as a tandem-compound unit, driving a generator of 30,000-kw. capacity and a 2000-kw. house generator. The low-pressure turbine, which is placed alongside the tandem element, is direct-connected to a 20,000-kw. generator. The exciter for the entire machine is also driven by the low-pressure element. Both units operate at 1800 r.p.m. At full load the high-pressure cylinder expands the steam from 550 lb. per sq. in. gage and 725 deg. F. to 132 lb. per sq. in. abs. with practically no superheat. The entire volume of steam is conducted to a reheat boiler and is returned to the intermediate-pressure cylinder at 122 lb. per sq. in. abs. and 700 deg. F. The steam is expanded to 16.5 lb. per sq. in. abs. and then passes directly to the low-pressure cylinder, from which it is exhausted at one inch of mercury absolute. Steam is extracted for feedwater heating from the intermediate cylinder at 50 and 17 lb. abs. and from the low-pressure cylinder at 4 lb. abs. The arrangement of the high-pressure and intermediate-pressure turbines is such that the steam flow is opposed so that one may neutralize the thrust of the other, eliminating the necessity for large balance pistons. The low-pressure element is of the double-flow type, the steam entering at the center of the rotor and flowing both ways into a 65,000-sq. ft. vertical condenser. Reaction blading is used throughout, and the last two rows of blades have a divided exhaust arrangement, a scheme for reducing the length of the low-pressure blades originally developed by K. Baumann of Manchester, England.

The 208,000-kw. steam turbine, Figs. 211 and 212, of the State Line

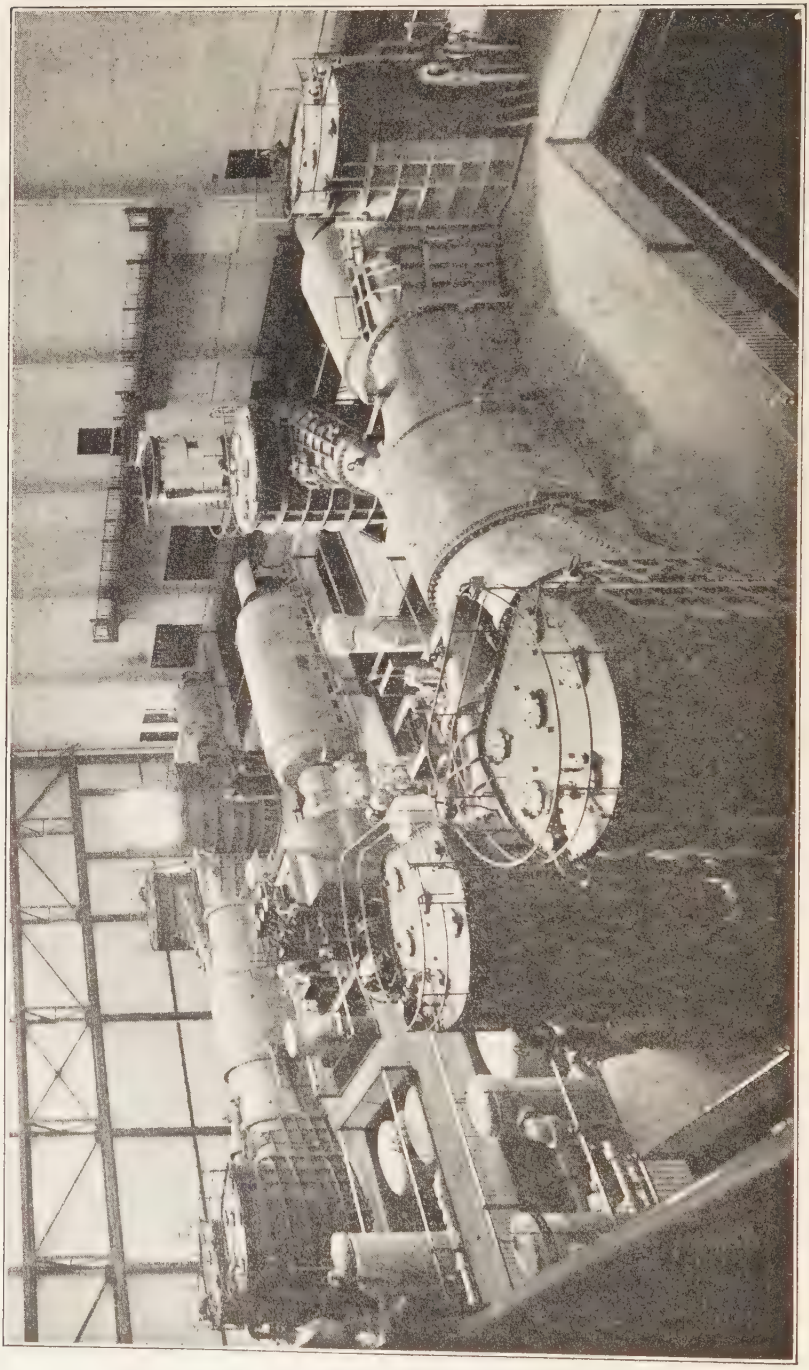


FIG. 211.—208,000-kw. triple-compound turbine-generator set, State Line Generating Station, Hammond, Indiana.

Generating Station, is a three-unit set operating at 1800 r.p.m. with one high-pressure (76,000 kw.) and two low-pressure cylinders (66,000 kw. each). The turbine is supplied with steam at 600 lb. per sq. in. gage, 730 deg. F. at the throttle; the entire steam flow passes through the single-flow high-pressure turbine, shown in the center of the illustrations. The exhaust steam is reheated at a pressure of 100 lb. per sq. in. gage to 500 deg. F. by live steam; half of the steam then goes to each of the low-pressure turbines. The low-pressure cylinder, Fig. 213, is of the double-flow type. Steam enters at *I*, divides and flows in two directions, thus practically eliminating end thrust and reducing the size of the

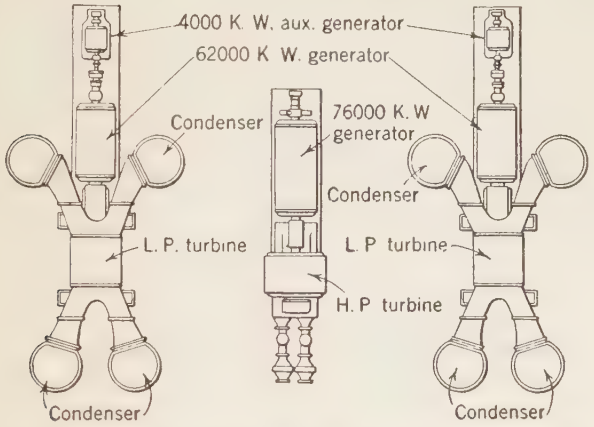


Fig. 212.—Plan of 208,000-kw. turbine-generator set showing arrangement and size of units.

rotor blades. The steam on leaving the last stage wheels flows horizontally to the right and left into exhaust hoods. The steam is again divided, Figs. 212 and 213, in the exhaust hoods of the low-pressure turbines, the total exhaust being disposed of to eight vertical condensers having a surface of 22,000 sq. ft. each. The stream-line exhaust hoods give a higher vacuum at the turbine wheels than at the condenser inlet, owing to the diffuser action.

Figure 214 shows a drawing of a 50,000-kw. vertical-compound turbine-generator set, in which the high-pressure turbine-generator, operating at 3600 r.p.m. and comprising 25 per cent of the total capacity of the set, is placed on top of the generator of the low-pressure element which operates at 1800 r.p.m. This arrangement conserves floor space, eliminates one foundation, and permits the use of one set of surface air coolers for both generators. Steam enters the high-pressure element at 1200 lb. per sq. in. pressure, 750 deg. F. total temperature; between the high-pressure and low-pressure elements the steam is

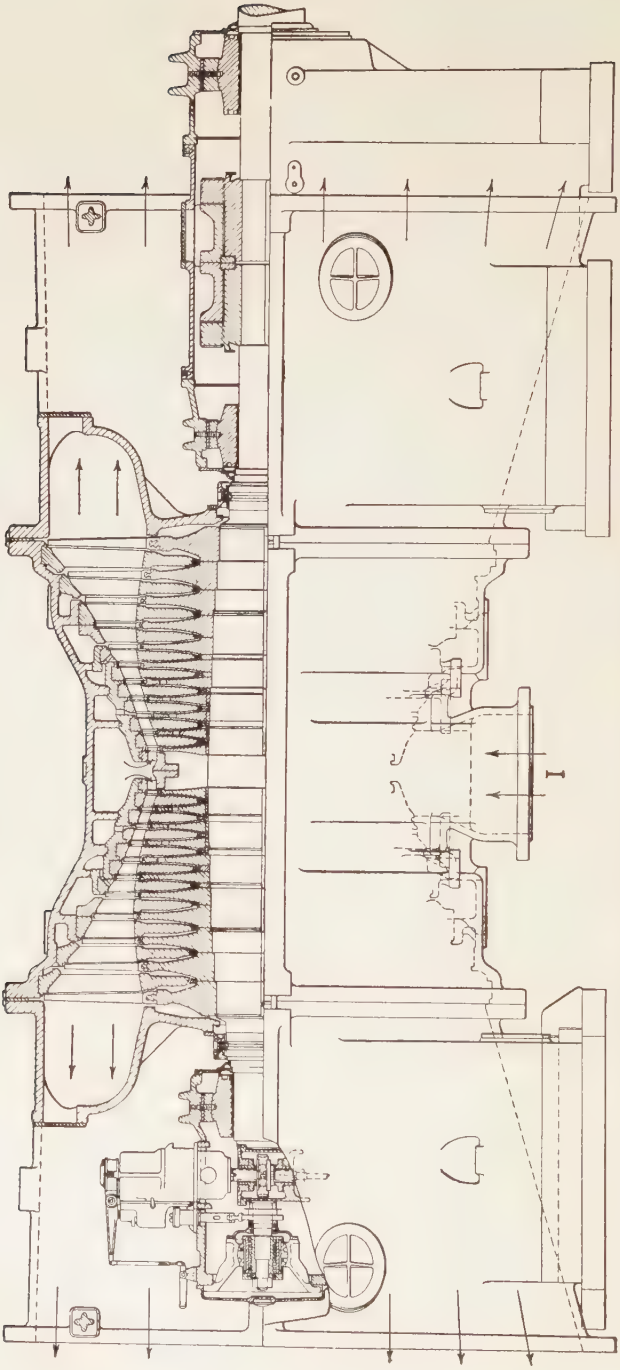


FIG. 213.—Low-pressure double-flow unit of State Line 208,000-kw. steam turbine.

returned to the boiler and reheated to 750 deg. F. The steam then expands through the low-pressure element to an absolute exhaust pressure of one inch of mercury. The largest unit of this type is a 110,000-kw. set installed at the River Rouge Plant of the Ford Motor Company; it differs slightly from Fig. 214 in that duplicate 55,000-kw. generators are used, and the high-pressure turbine which operates at 1800 r.p.m. is directly over the 1800 r.p.m. double-flow low-pressure turbine. Steam conditions are: at the inlet, 1200 lb. per sq. in. gage pressure and 725 deg. F.; reheat to 550 deg. F. at 80 lb. per sq. in. gage pressure; and exhaust at a pressure of one inch of mercury absolute.

249. Back-pressure Turbines.—The name of back-pressure turbine is usually given to high-pressure turbines that operate non-condensing. One application of turbines of this type is their use when it is desired

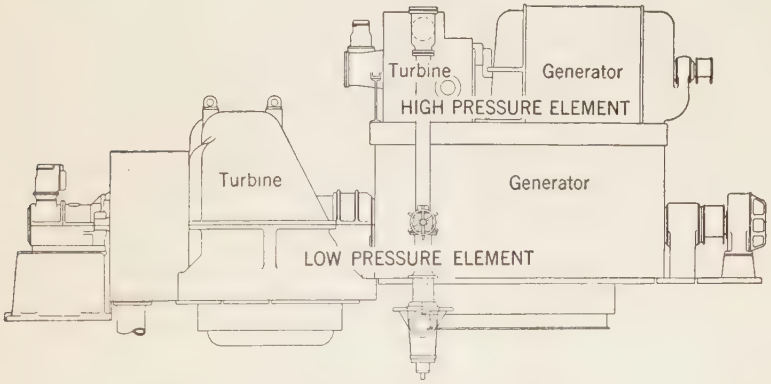


Fig. 214.—Vertical-compound turbine-generator set.

to increase the generating capacity of a plant and at the same time maintain the existing steam equipment which operates at a relative low or medium steam pressure. Simple non-condensing steam turbines are also used when all or practically all of the exhaust steam can be employed for process heating and the heating of buildings, or when the cost of fuel or steam is such that the additional investment required for condensing equipment or a more intricate turbine is not justified. This type of turbine is generally employed in the smaller sizes where the additional cost of a more elaborate design of turbine would exceed the saving resulting from the possible lower steam consumption. Back-pressure turbines may be built either in the single-stage or in the multistage type. When low steam consumption is desired the multistage is used, as this type is more efficient for average conditions. This is especially true in units of 100 kw. and above; the field for a highly

efficient turbine in the smaller sizes does not ordinarily justify the development of a turbine of the multistage type.

The steam pressure required for heating or industrial purposes may range from atmospheric to 100 lb. per sq. in. gage pressure or more. However, the steam consumption of the turbine for a given load increases with the increase of back pressure. Some industrial plants obtain their low-pressure steam by passing high-pressure steam through reducing valves. Back-pressure turbines serve as rotary pressure-reducing valves whereby power is delivered as a by-product in the production of low-pressure steam. The low-pressure steam is delivered to the exhaust, clean and suitable for any kind of heating. As only a small portion of the available heat is converted into power by the turbine, the remaining heat in the steam is still available for process heating or the heating of buildings. Paper mills, oil refineries, soap factories, rubber mills, tanneries, textile mills, sugar refineries, chemical plants, canneries, laundries, etc., make use of low-pressure steam in some phase of their processes.

The problem of governing is particularly simple, if the turbine is operating in parallel with other turbines which take care of governing the station output. The steam-inlet valve is then controlled by a pressure governor which regulates to give a constant back pressure and allows as much steam to pass through the turbine as is needed for heating. A speed governor is also provided, which functions when the total output of the station falls below the maximum output of the back-pressure turbine. The addition of an extra valve for sending live steam into the exhaust main is desirable. Such an arrangement acts independently of the opening of the turbine steam-inlet valve, when the exhaust steam pressure has attained a certain minimum value. There are installations in which the demand for heating steam is occasionally larger than the quantity of steam passing through the turbine at maximum output. On the other hand, when no heating steam is required while power is needed, the turbine should be capable of operating with atmospheric exhaust.

Since back-pressure turbines operate as reducing valves and utilize only 5 to 10 per cent of the total heat in the steam, power is produced practically as a by-product and at a minimum of cost. A 1000-kw. turbine operating at 165 lb. per sq. in. abs. initial steam pressure (dry saturated) and 30 lb. per sq. in. abs. back pressure, the exhaust steam being used for heating purposes, showed steam consumptions, Fig. 215, as follows: full load, 49 lb. per kw-hr.; three-fourths load, 49.5 lb. per kw-hr.; half-load, 52 lb. per kw-hr.; and one-fourth load, 60 lb. per kw-hr.

250. Extraction Turbines.—The field of application of the back-pressure turbine is limited, as in some industries the demand for low-

pressure steam is not constant, that is, the demand for heating may have no direct relation to the demand for electrical energy. In such cases, in order to obtain greater flexibility and better efficiency of the combined power and heating plant, an *extraction turbine* is used. An extraction or bleeder turbine is a high-pressure condensing turbine in which provision is made for the extraction of the amount of steam required for heating or other industrial purposes, from an intermediate stage of the turbine. Partially expanded steam in any quantity from

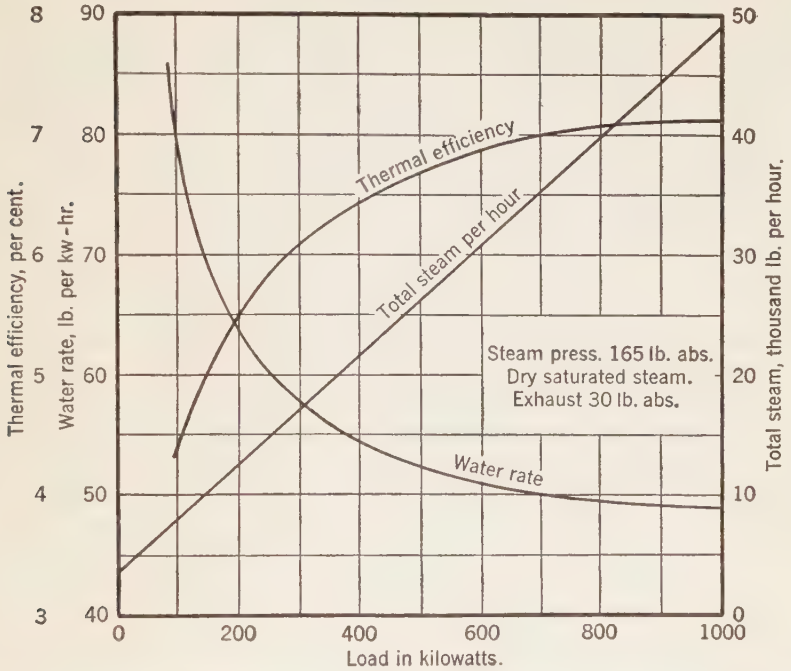


FIG. 215.—Performance curves of a 1000-kw. non-condensing (back pressure) steam turbine.

zero to 100 per cent of the entering steam may be removed from a turbine even though the demand for bled steam is variable. Suitable automatic or hand-operated valves are used to control the quantity of steam so extracted.

Figure 216 shows a 3000-kw. extraction turbine with a diaphragm or partition between the high-pressure impulse and the low-pressure reaction stages of the turbine. Ports through this partition are opened and closed by the governor-controlled butterfly valve. The movements of the butterfly valve and the diaphragm are so correlated that as the ports of the partition are opening the butterfly valve is closing. When

extracted steam is not required, the governor closes the butterfly valve and opens the partition ports; the turbine then operates as a high-

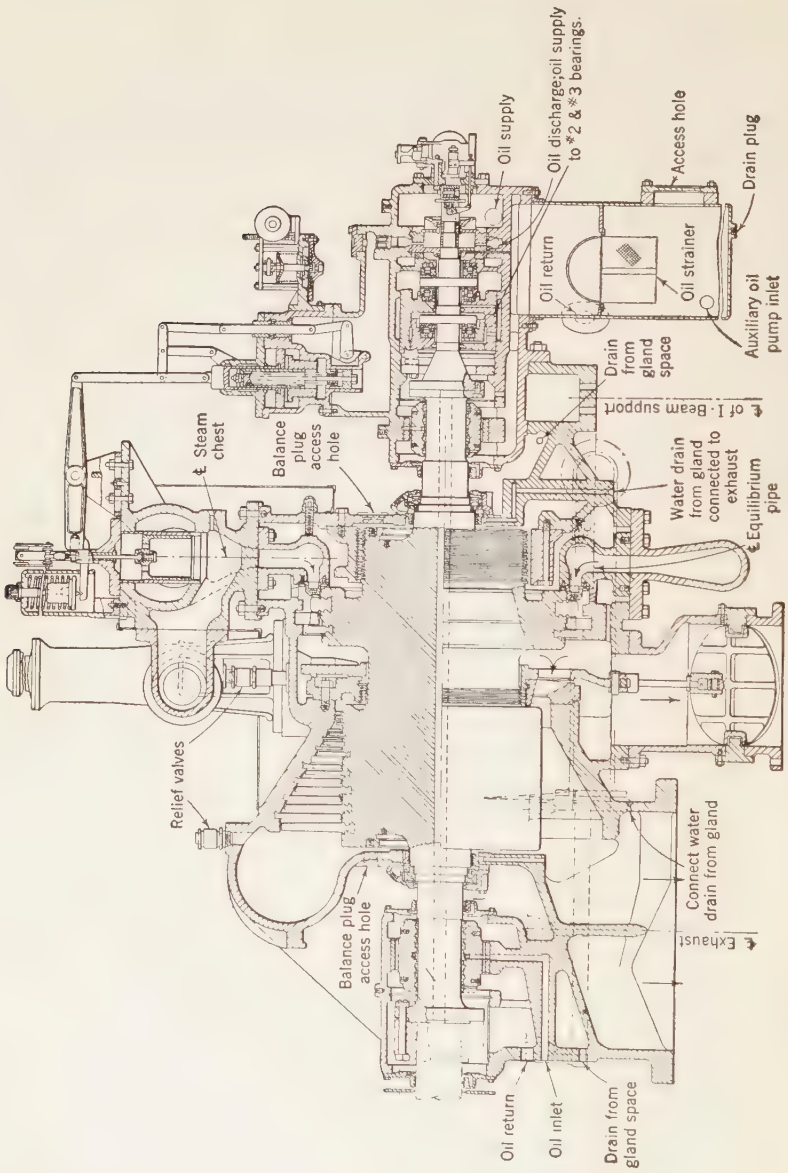


FIG. 216.—Westinghouse 3000-kw. combined impulse and reaction turbine, extraction type.

pressure condensing unit. When the ports are partially closed, the pressure in front of the diaphragm rises and more steam is discharged into the bleeder pipe connection through the partially opened butterfly

valve. When the partition ports are closed and the turbine is operating straight non-condensing, the low-pressure stages are revolving in a partial vacuum, thus relieving the turbine of the drag which the inactive stages would otherwise cause.

Provision has been made on all recent large turbines for the extraction of steam at various stages for use in feedwater evaporators, heaters, and deaerators. This method, for the more complete utilization of the heat of the steam, is known as the regenerative cycle. Turbines have been bled at as many as four points, and such procedure results in substantial reduction in the heat consumption of the plant as a whole. At the Richmond Station of the Philadelphia Electric Company, a 50,000-kw. tandem-compound impulse turbine with a total of 20 stages receives steam at 390 lb. per sq. in. abs. and a temperature of 675 deg. F. Steam from the tenth stage at 130 lb. per sq. in abs. is taken to a feedwater evaporator; steam from the twelfth stage at 79 lb. per sq. in. abs. goes to feedwater heaters and evaporators. Steam from the fifteenth stage at 28 lb. per sq. in abs. goes to feedwater heaters and a deaerator, and from the eighteenth stage at 5.6 lb. per sq. in. abs. to a feedwater heater.

The curves in Fig. 217 show the influence of the amount of steam extracted on the hourly steam consumption of a 1000-kw. extraction turbine. The unit water-rate curve for condensing operation is also shown. The unit water rate at any load, for any amount of extraction, and the heat content of the bled steam may be readily calculated from the diagram. Enter the curves at any partial load, select 500 kw., move vertically to the diagonal line of the desired amount of steam to be bled (use 15,000 lb. per hour), then horizontally to the pounds of steam required at the throttle per hour, 20,000 lb. in this case. Therefore, the unit water rate is 40 lb. per kw-hr. The work done by 20,000 lb. of steam per hour, non-condensing, is found from the diagram to be about 320 kw. with a corresponding unit water rate of 62.5 lb. per kw-hr. If the upper diagonal line were reached on the vertical move, the turbine would be operating non-condensing, and this would be the maximum amount of steam to flow per hour at this load. At a 500-kw. load operating condensing, no steam bled, the steam required would be only 10,000 lb. per hr. or 20 lb. of steam per kw-hr.

251. Low-pressure Steam Turbines.—The reciprocating steam engine utilizes economically a greater amount of the available energy of the steam at high pressures than at low pressures. The turbine, on the other hand, has the advantage at low pressures and in cases where large ratios of expansion are used. About the same amount of energy is available for doing work in expanding from a medium boiler pressure down to that of the atmosphere, as there is in expanding from atmos-

pheric pressure to that of a good vacuum. However, the amount of expansion is much larger in the second case. To utilize this expansion at a low pressure the steam-engine cylinder would be very large and a large cylinder loss would occur. With the turbine the reverse is true, that is, the turbine suffers relatively less loss at a low pressure than at a high pressure.

The ability of the turbine to handle low-pressure steam to advantage has given rise to the use of *low-pressure turbines* in many different ways.

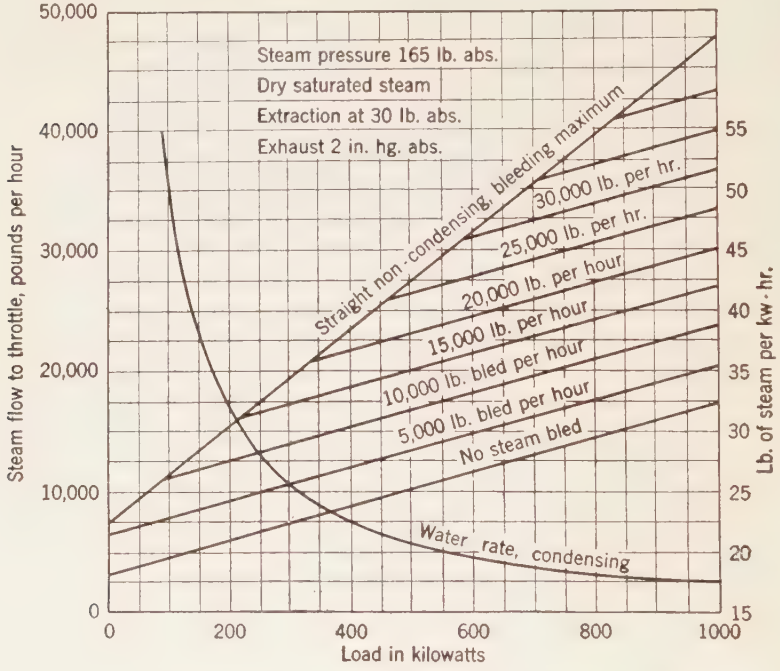


Fig. 217.—Typical steam consumption curves of 1000-kw. extraction turbine.

As examples, the exhaust of hoisting engines, steam hammers, and other apparatus, commonly exhausting at or near atmospheric pressure, may be led to one or more low-pressure turbines. Further expansion in the turbines permits the recovery of a large amount of power from what would otherwise be waste steam. On account of the intermittent operation of hoisting engines, steam hammers, and rolling-mill (steel plant) engines, the low-pressure turbines do not receive steam continuously. This difficulty may be overcome by using a *steam accumulator*, which consists of a large (closed) cylindrical vessel interposed between the rolling-mill engine and the low-pressure turbine. About 90 per cent of the total capacity of the storage vessel is occupied by

water, which large quantity is required to readily condense the entering exhaust steam. During the lowered pressure existent at the intervals of reduced steam supply, vaporization of the stored water at the expense of the contained heat augments the steam supply and assures continuity of steam flow to the turbines. The steam accumulator (thermal storage) system is also applicable where a variable supply of steam is required, as in the pulp, paper, chemical, refining, food, textile, and rubber industries.

252. Mixed-pressure Turbines.—A mixed-pressure turbine may be regarded as a low-pressure turbine with a number of high-pressure stages. The latter enable additional high-pressure steam to be used when the maximum available amount of low-pressure steam is insufficient to take care of the power requirements. A common type of mixed-pressure turbine consists of an impulse section in which the heat drop from boiler pressure to about atmospheric pressure is absorbed, and the remaining available energy is removed by expansion in low-pressure reaction blading. The valves under the control of the governor are adjusted so that no high-pressure steam is admitted until the valves on the low-pressure steam line are wide open. In some mixed-pressure turbines the high-pressure section, if of a simple disk construction, is frequently made removable so that the "windage" loss (about 1 to 3 per cent), due to rotation when not in use, can be eliminated. Mixed-pressure turbines are being built in sizes up to 10,000 kw. with a steam consumption of from 20 to 35 lb. per kw. per hr., depending upon the size, initial steam conditions, per cent of rating developed, and the turbine back pressure. This type of turbine has practically supplanted the straight low-pressure design in the modern industrial plant.

253. Energy Losses in Steam Turbines.—The most important energy losses that occur in steam-turbine operation are: steam leakage, wheel and blade rotation, residual velocity of the steam leaving the blades, heat content of the exhaust steam, radiation, and friction.

To prevent leakage of steam out of or air into the turbine, where the shaft passes through the ends of the casing, either a ring-packed or water-sealed gland or a steam-packed labyrinth is used. In impulse turbines there is steam leakage between the diaphragms and the shaft; in reaction turbines leakage occurs over the ends of the blades through the radial clearance passages. This leakage loss averages about 10 per cent in most turbines. There is: a frictional resistance between the steam and the sides of a nozzle, a loss due to impact as the steam enters the moving blades, eddying and fluid friction in the blades. The windage loss due to the friction of the turbine wheel and blades revolving in steam may in some cases be very great. Tests have shown that this loss depends upon the density and quality of the steam and the speed

of the wheel. If dry saturated steam is used it will be wet after expansion and the presence of the moisture will greatly increase the friction. The greater economy which results from the use of superheated steam is doubtless due to the reduced frictional resistance resulting from the absence of water in the steam. Radiation and conduction losses are usually very small because of effective covering of the turbine casing and piping with heat-insulating material. Other losses include bearing friction, friction of the gears (if any are used), and power for the governor and oil pump.

254. Efficiency and Economy of Steam Turbines.—As the steam pressure, steam temperature, and capacity of turbines have been

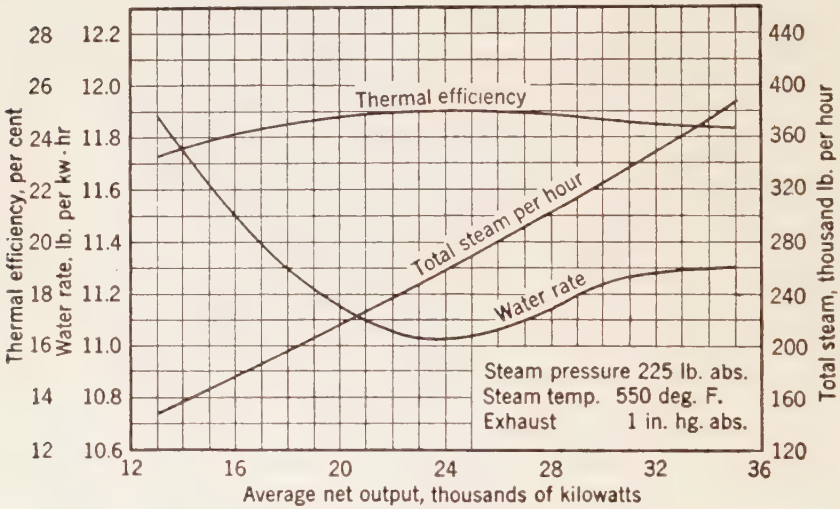


FIG. 218.—Performance curves of a 30,000-kw. condensing steam turbine.

increasing, the heat used per kilowatt-hour has decreased until the thermal efficiency of the modern large unit is about 30 per cent. The steam turbine is well adapted to the driving of electric generators of the alternating-current type as the desirable speeds of the two machines are the same. Such units can be constructed in large capacities with high efficiencies and are economical in the use of steam at a comparatively low cost per kilowatt. These advantages have resulted in bringing the steam turbine into almost universal use where steam is used for producing power with units in excess of 1000-kw. capacity.

General comparisons of the steam consumption of turbines and reciprocating engines are difficult because of the diversity of operating conditions. However, the steam consumption of the better varieties of the two types is of the same order for comparable conditions, with the

advantage probably slightly in favor of reciprocating engines in the smaller sizes and for non-condensing service, but decidedly in favor of the turbine in the larger sizes for condensing service. The steam turbine operates on the complete expansion cycle; the reciprocating engine operates on a cycle with incomplete expansion. The turbine therefore has a certain theoretical advantage because its cycle is such as to convert into work a greater amount of heat per pound of steam used between given upper and lower pressures.

Figure 218 shows the results of a test on a 30,000-kw. condensing turbine, with the least steam consumption and the highest thermal efficiency at 80 per cent of the rated load. However, the economy and efficiency of this turbine do not vary greatly between 18,000- and 35,000-kw. output, thus insuring good economy over a wide range of load. Typical performance curves for back-pressure and extraction turbines are illustrated in Figs. 215 and 217, respectively. These curves show that the efficiency and economy of small turbines, condensing and non-condensing, are of about the same order as for good steam engines; compare the above curves with Fig. 172. But the thermal efficiency and steam economy are only two of the many factors entering into the selection of a prime mover and the ultimate cost of power. The compound steam engine is limited in size to about 5000 kw., and the speed seldom exceeds 300 r.p.m. Other comparative factors are given in the last paragraph of Art. 236, page 300.

255. Trends in Steam Turbine Development.—Present trends of steam turbine development are toward: the use of higher steam temperatures and pressures; increased capacities at the various speeds and increased output from a single cylinder; the use of better materials for turbine parts, and particularly for blading; higher efficiency over a wide range of load; adjustment of the turbine more closely to the particular requirements of service; and the development of binary fluids for use in turbines, as explained in Art. 257. The effect will be the production of larger, simpler, more reliable, and more economical turbines. The use of turbines of larger rated capacity results in several economic gains per kilowatt; such as less floor space, decreased weight, and lower first cost. Efficiencies tend to increase with higher turbine capacities, though at a lesser rate in the largest units. Reduction in weight and higher efficiency over lower speed conditions (1200 or 1800 r.p.m.) are leading to a wider use of large turbines operating at 3600 r.p.m.

Blade tip speeds up to 1200 ft. per sec. and high moisture content in the exhaust steam from the use of large heat drops at high efficiency are demanding improved materials. Stainless steels at present seem best suited for long blades. A steel with about 12 per cent chromium

and 0.10 per cent carbon has shown promise for such service. To resist erosion on the back of the inlet edge of the blade, properly heat-treated stainless steels and some nickel alloys seem best. Plating the edges of these blades with chromium, or spraying coatings of tantalum, using sheaths of Hecla and other metals, coatings of stellite, and nitriding, also appear to aid resistance to erosion. With materials now available and parts designed so that the working stresses will lead to a rate of *creep* [1] of 0.01 per cent per year, turbines can be built for steam temperatures up to 1000 deg. F.

Steam conditions in future stations probably will be standardized as follows: without reheating, 400 lb. per sq. in. and 750 deg. F., or 700 lb. per sq. in. and 850 deg. F.; with one stage reheating, 1200 to 1400 lb. per sq. in. and 750 deg. F., or 2000 to 2500 lb. per sq. in. and 900 deg. F. to 1000 deg. F. In the latter case, two stages of reheat could be used if high efficiency were desired. Although these considerations apply, in general, to large central stations, the gains from increased temperatures and pressures are equally applicable to industrial plants. In fact, when such plants operate non-condensing by exhausting against back pressure into a service main, the proportional gains from using high pressure and high superheat are larger than in condensing stations.

Improved turbine efficiency and economy may be expected to result from lower exhaust losses, from reduction in the amount of moisture in the exhaust stages, from improved blading in the high-pressure section, and from a better combination of impulse and reaction effects in all blading. The probable limiting performance of a steam turbine, within the bounds of present knowledge, is a station heat rate of 9500 B.t.u. per kw-hr. net output. This figure may be expected for a large plant with steam at 2500 lb. per sq. in. and 1000 deg. F., having two stages of reheating to the same temperature, and with six-stage steam extraction.

256. Reheating and Regenerative Cycles.—A study of the performances of generating stations employing pressures of 400 lb. per sq. in. gage and temperatures of 750 deg. F. shows that there is little net advantage in exceeding these limits for turbines operating on the straight Rankine cycle. With the practical temperatures limited to 850 to 900 deg. F., the increased range of expansion for initial pressures over 400 lb. per sq. in. causes the steam to become saturated too soon to permit the lower stages of the turbine to function advantageously. By reheating the steam after partial expansion and by extracting steam from the turbine at several points for feedwater heating, the gain in efficiency is such as to warrant the adoption of pressures up to 2500 lb.

[1] Creep may be defined as the very slow expansion of a metal due to continued high temperature and pressure.

per sq. in., as outlined in Art. 255. This intermediate reheating of the steam between stages and bleeding for heating purposes puts the operation of the turbine in a cycle other than the Rankine cycle.

Because of the impracticability of the Carnot and the limitations of the Rankine cycles, other heat cycles have been adopted which offer higher efficiencies than the Rankine. These improved cycles include: (1) the reheating, (2) the regenerative, (3) the reheating-regenerative, and (4) the binary-fluid cycles.

In the *reheating cycle* an original superheated steam supply is partially expanded through the high-pressure portion of the turbine and is then returned to the boiler room and resuperheated, after which the expansion is completed through the low-pressure portion of the turbine. The ideal temperature-entropy diagram for single-stage reheating is shown in Fig. 219. The portion of the curve *0123* is the same as for the Rankine cycle. From *3* the steam expands adiabatically to *4*; at *4* it is reheated at constant pressure to some point *5*, after which a second expansion takes place from *5* to *6*. From *6* to *0*, condensation of exhaust takes place at constant pressure just as in the Rankine cycle. The work done by a pound of steam in expanding to the reheating point *4* is $(h_3 - h_4)$, and in expanding from *5* to *6* is $(h_5 - h_6)$; the initial heat supplied is $(h_3 - h_{f,0})$ and that during the process of reheating from *4* to *5* $(h_5 - h_4)$. Total work done $= (h_3 - h_4) + (h_5 - h_6)$; total heat supplied $= (h_3 - h_{f,0}) + (h_5 - h_4)$, hence

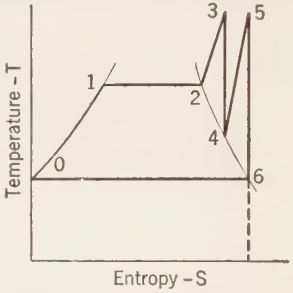


FIG. 219.—Single-stage reheating cycle.

$$\text{Reheating Cycle Eff.} = \frac{(h_3 - h_4) + (h_5 - h_6)}{(h_3 - h_4) + (h_5 - h_{f,0})}; \qquad (132)$$

in which h = total heat of the steam, B.t.u. per lb., and h_f = heat of the liquid, B.t.u. per lb.; above 32 deg. F. for the state indicated by the subscript in each case.

The reheating arrangement offers a method whereby advantage is taken of the higher efficiency possibilities which result inherently from increase of the average temperature at which the working medium receives additional heat. Practical benefit of reheating arises from the evident deferring or avoidance of condensation during expansion. The appearance of moisture during expansion has a particularly serious influence on the fluid friction losses and the efficiency of the steam turbine. The cycle has its greatest utility in those more modern plants

which employ pressures of 1000 lb. or more, thus the resuperheating may be done at a steam pressure of 300 lb. or more when the steam still is of moderate specific volume, and large return piping and large reheaters are avoided. The greatest disadvantage of the arrangement lies in the necessity for repiping the steam to the boiler room and providing a second superheater.[2]

In the *regenerative cycle* the condensate from the turbine is passed through a series of feedwater heaters where the heating medium is steam which has been extracted from several pressure stages of the turbine. With an infinite number of heaters the temperature of the feedwater could be increased to the boiler temperature and the efficiency would be that of the Carnot cycle. This is true only for saturated steam, since with superheated steam the maximum theoretical temperature of the condensate can never exceed that of saturated steam corresponding to steam at initial or throttle conditions and therefore the efficiency will be less than that of the Carnot cycle.

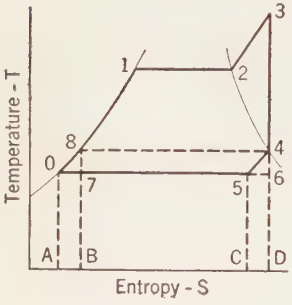


Fig. 220.—Regenerative (extraction) cycle.

Figure 220 shows the temperature-entropy diagram of an ideal regenerative cycle with an infinite number of stages in which the maximum temperature of the feedwater is taken as that of the liquid at a pressure corresponding to the point in the expansion line where saturation begins, i.e., where superheat disappears. In this particular cycle, therefore, no superheated steam is extracted. The path *01234* is the same as in the conventional Rankine cycle. Point *4* is the saturation point on the expansion line and represents the initial extraction point. If the steam had been expanded adiabatically, the path would be represented by *46*, and *0123460* would have been the heat available. If the extracted steam and the condensate of the turbine were mixed together, the history of the state of the steam in the turbine would be represented by the line *45*. The line *50* represents the rejection of heat to the condenser. The line *45* is drawn parallel to *08* since the heat absorbed by the condensate *A08B* is equal to that absorbed by the extracted steam *C54D*. The heat converted to work is represented by the area

[2] To avoid this disadvantage several modern generating stations are reheating with high-pressure superheated steam which is taken directly from the main steam supply and delivered to a heater through which the medium-pressure steam is passed. Although this simplifies the installation, the thermodynamic gain of resuperheating is obviated by the thermal irreversibility of the heat transfer in the heater and the inability to remove all the "available" energy from the high-pressure steam.

0123450 = area *A0123D* − area *A084D* + area *7846* (= area *0845*); or $(h_3 - h_4) + (T_4 - T_0)(s_3 - s_{f.0})$, which reduces to $(h_3 - h_{f.8}) - T_0(s_3 - s_{f.8})$. The net heat supplied is area *B7812345C* = $(h_3 - h_{f.8})$.

Therefore

$$\text{Regen. Cycle Eff.} = \frac{(h_3 - h_{f.8}) - T_0(s_3 - s_{f.8})}{(h_3 - h_{f.8})}; \qquad (133)$$

in which h = total heat of steam, B.t.u. per lb., h_f = heat of liquid, B.t.u. per lb., T = absolute temperature, deg. F., s = total entropy per lb., and s_f = entropy of liquid per lb.; for the state indicated by the subscript in each case.

In extraction feedwater heating it is to be noted that the gain over the straight Rankine cycle is accomplished, not by extending the limits of the cycle as in the reheating cycle, but by a redistribution of the heat within the original limits. For this reason, the plan has received primary consideration, as well as the fact that it is perhaps the most easily embodied in an actual steam power plant.

The *reheating-regenerative cycle*, Fig. 112, is being used in some of the recent installations of multicylinder turbines with reheating between cylinders and extraction from any or all of the cylinders. The thermal efficiency of this combined cycle may be calculated by taking each step in the cycle and analyzing as outlined in the previous paragraphs of this article.

257. Binary Vapor Cycle: Mercury and Steam.—The thermal efficiency of any heat engine may be increased by extending the range of heat availability of the working medium. For each fluid there is a pressure and temperature range beyond which it is impracticable to go because of the physical limitations of the materials employed in generating heat at the higher level, and because of the physical properties of the working fluid and cooling media at the lower level. In the high-temperature region, steam carries with it an unreasonably high pressure; and in the low-temperature region, a troublesomely large specific volume. Mercury boils under standard atmospheric pressure at 675 deg. F., and under a pressure of one pound per square inch absolute at 458 deg. F. The corresponding temperatures for water are 212 deg. F. and 102 deg. F., respectively, and for methyl alcohol 150 deg. F. and 50 deg. F., respectively.

It is possible and practicable to utilize certain of the advantages of a fluid and in some degree suppress its disadvantages by the joint use of two or more fluids in a successive arrangement of cycles. In such an arrangement, which would be known as a multiple-fluid cycle, advan-

tageous pressure-temperature characteristics may be utilized by employing a fluid of relatively low vapor pressure in the higher-temperature section of the composite cycle and a fluid of high vapor pressure in the lower-temperature section. A compound cycle of this type which uses the two fluids mercury and water has been in commercial service since February 4, 1930, at the South Meadow Station of the Hartford Electric Light Company, Hartford, Connecticut.[3]

Experience with the mercury-steam process devised by Dr. W. L. R. Emmet of the General Electric Company, as installed at Hartford, proves conclusively that this method of power generation is sound in practice as well as in theory. The general scheme of arrangement of this pioneer commercial installation is shown by Fig. 221; the energy

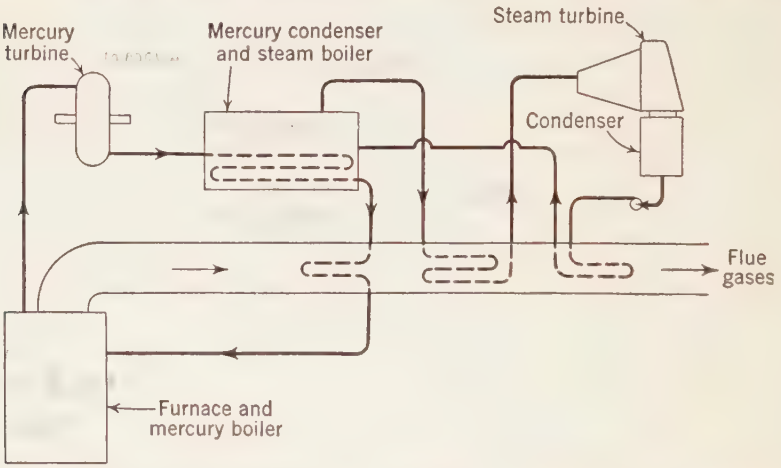


FIG. 221.—Schematic drawing showing arrangement of mercury-steam installation.

rejected from the condensing mercury forms the energy source for vaporizing the water. The mercury boiler, drop-tube type, is fired with pulverized coal and generates mercury vapor at 70 lb. per sq. in. gage and 884 deg. F.; the five-stage mercury turbine and accessory equipment deliver 10,000 kw. in electrical energy and 129,000 lb. of steam per hour. The mercury vapor is expanded to 1.5 in. of mercury absolute, at which pressure the temperature is 440 deg. F. Latent heat of the exhaust mercury is then transferred to feedwater at 275 deg. F. to make steam at a pressure of 280 lb. per sq. in. gage in a heat exchanger

[3] "Mercury-Vapor Unit Operates Successfully at South Meadow," by James Orr; *Power*, July 1, 1930. "Mercury-Vapor Unit a Commercial Success," by T. H. Soren; *Electrical World*, July 5, 1930. "10,000-Kilowatt Mercury Turbine Plant," *London Engineering*, January 16, 1931. "The Emmet Mercury-Vapor Process," by B. P. Coulson, Jr.; *General Electric Company*, GED–497, February, 1931.

which is both a mercury condenser and a steam generator. The steam thus formed, after being superheated to 735 deg. F., is utilized in a steam turbine and expands to the condenser exhaust pressure.

Two additional 20,000-kw. mercury turbines are now (1933) under construction. Ole, at Schenectady, New York, will deliver 20,000 kw. to the New York Power and Light Company and will send 325,000 lb. of steam per hour to the General Electric Company Plant. The other, at the Kearny Station of the Public Service Electric and Gas Company of New Jersey, will deliver steam to turbines in which 33,000 kw. will be generated, thus making, with the 20,000 kw. from the mercury turbine, a total of 53,000 kw. for the combined mercury-steam unit. Table XIX gives details of the three mercury-steam plants discussed above.

On account of the low nozzle velocity of the mercury vapor, bucket speeds are about one-third that of steam. All wheel clearances are relatively large. The turbine and exhaust casing is in one piece branching to two exhaust openings, one on each side, which serve to lead the exhaust mercury vapor to the two condenser-boilers.

TABLE XIX

Comparative Operating Conditions of Mercury-Steam Plants

	Hartford	Kearny	Schenectady
Load on mercury turbine, kw	10,000	20,000	20,000
Speed of mercury turbine, r.p.m	720	900	900
Mercury-vapor pressure at turbine, lb. gage	70	125	125
Mercury-vapor temperature at turbine, deg. F	884	958	958
Pressure in mercury condenser, in. mercury abs	1.5	3	3
Temperature of mercury in condenser, deg. F	440	485	485
Total steam from unit, lb. per hr	129,000	325,000	325,000
Steam pressure, lb. gage	280	365	400
Steam temperature after superheating, deg. F	735	750	760

Commercial operation over a period of several days with a continuous load of 9500 kw. on the mercury turbine-generator at Hartford has given a net economy of 9800 B.t.u. per kw-hr., 34.8 per cent thermal efficiency; and over a period of nine months 10,180 B.t.u. per kw-hr. The expected fuel rate is 9500 B.t.u. per kw-hr., 35.9 per cent thermal

efficiency, with the two higher initial temperature (20,000 kw.) mercury-steam plants. Considering the mercury vapor-steam cycle with mercury-cooled walls instead of the water-wall furnace used with pulverized-coal installations, a heat rate of 8800 B.t.u. may be expected; this figure is based on the performance of the Hartford installation and the using of an efficient 400 lb. per sq. in. pressure steam turbine from which steam will be extracted to heat the feedwater to 350 deg. F. The Kearny and Schenectady furnaces have mercury walls in the upper half and water walls in the lower half.

Operating experiences with the Hartford installation of the mercury-steam cycle indicate that little maintenance should be required on the mercury boiler. The external cleaning of the tubes is easy, as they are short and in practically a vertical position. The furnace maintenance is the same as for any pulverized-coal-fired steam boiler. The return of mercury to the boiler by gravity eliminates boiler feed pumps, valves, and regulators. As regards the mercury turbine, operating experiences at Hartford show no excessive maintenance associated with operation at the high temperatures involved. Wear of the turbine blades has not been serious.

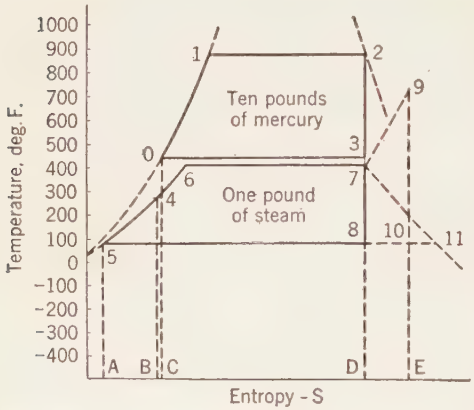


FIG. 222.—Temperature-entropy diagram of mercury-steam cycle.

Considering the mercury condenser: Tube failures caused by erosion, corrosion, or electrolysis have not occurred. As a result of the mercury vapor generating approximately half of the station output, the circulating water requirements are reduced by half, with a corresponding reduction in the sizes of the intake and discharge tunnel and water-screening equipment.

Regarding the condenser as a steam boiler: There is no burning or distorting of tubes as in a boiler with a fire under it. Irrespective of the amount of scale-forming solids in the feedwater, the condenser-boiler would require less care than an ordinary steam boiler.

Figure 222 shows temperature-entropy diagrams for 10 lb. of mercury and for 1 lb. of water. These are about the requisite proportions between the rates of mercury and water circulation in the cycle, because the heat delivered in condensing one pound of the mercury vapor after

expansion through the mercury turbine is about one-tenth of that required to evaporate one pound of water. In Fig. 222 the mercury turbine cycle is *0123* and the steam turbine cycle *5679, 10*. The primary heat received by the water along the line *467* is equal to that given up by the condensing mercury along *30*, and the area *B467D* is equal to the area *C03D*. Point *4* represents the temperature, 275 deg. F. in this particular cycle, of the feedwater to the mercury condenser-steam boiler. The superheat of the steam represented by the area *D79E* under line *79* is added in the steam superheater installed at the top of the mercury boiler setting.

PROBLEMS

1. A turbine-generator delivered one kilowatt-hour for every 30 lb. of steam supplied. The steam at an initial pressure of 150 lb. per sq. in. abs. was superheated 70 deg. F. The back pressure was 15 lb. per sq. in. abs. Find the Rankine cycle efficiency and the thermal efficiency.

2. Find the Rankine cycle efficiency of a turbine with a constant exhaust pressure of 1.5 in. mercury abs. for supply pressures of 120, 250, 400, 700, 900, 1200, 1800, 2500, and 3226 lb. abs., if dry saturated steam was used in each case? Tabulate results and plot a curve to show how the efficiency of the Rankine cycle varies with the initial pressure.

3. Steam expanded adiabatically in a frictionless nozzle from an initial pressure of 200 lb. per sq. in. abs. and a temperature of 450 deg. F., to a back pressure of 15 lb. per sq. in. abs. Weight of steam discharged, 2700 lb. per hr. (*a*) Find the theoretical velocity of the jet of steam at the throat and exit of the nozzle. (*b*) What was the quality of the steam at the throat and the exit? (*c*) Determine the diameter of the nozzle at the throat and the exit; also length of nozzle in inches.

4. A turbine-generator had an output of 2000 kw. and a water rate of 18 lb. per kw-hr. The steam entered at 250 lb. per sq. in abs. and 100 deg. F. superheat. Find the diameter at the throat of each of *four* nozzles if the pressure drop was 150 lb. per sq. in. abs.

5. Determine the weight of steam, in pounds per hour, that can be delivered by a nozzle whose throat diameter is 0.5 in., when dry steam is supplied at a pressure of 200 lb. per sq. in abs.

6. Design and draw a nozzle to expand 1200 lb. of steam per hour from an initial pressure of 250 lb. per sq. in. abs. to a back pressure of 20 lb. per sq. in. abs. for steam that is: (*a*) superheated 150 deg. F., (*b*) dry and saturated, and (*c*) wet with a quality of 97 per cent. Assume constant entropy in each case. Give all dimensions in inches.

7. Solve problem 6 assuming that in each case there is a 15 per cent loss in the nozzle, due to friction.

8. Assume that steam with a quality of 99 per cent is passed through a theoretically perfect nozzle at the rate of $\frac{1}{2}$ lb. per sec. and is expanded from a pressure of 190 lb. per sq. in abs. to 16 lb. per sq. in. abs. (*a*) Find the quality of the steam at throat and exit of the nozzle. (*b*) Calculate the velocity of the steam at the throat and the exit. (*c*) Calculate the volume of the steam at the throat and the exit. (*d*) Determine the diameter of the nozzle at the throat and the exit, also the length of the nozzle in inches. (*e*) Draw a full-size sketch of the nozzle.

9. (*a*) If turbine blades transformed 80 per cent of the issuing steam-jet velocity of the nozzle given in problem 8 into useful work, calculate the horsepower developed in the rotor of the turbine. (*b*) For the production of maximum work, what should be the speed of the turbine rotor if the distance from the center of the rotor to the center of the exit of the steam nozzle is 14 in.?

10. A reciprocating engine received steam at 150 lb. per sq. in. abs. and 98 per cent quality, and exhausted at a pressure of 15 lb. per sq. in. abs. If a low-pressure turbine were added and the exhaust pressure lowered to 2 in. of mercury abs., what would be the theoretical increase in the available heat in per cent of that initially available to the engine?

11. Plot a complete expansion cycle, using absolute pressure as ordinates and energy available in B.t.u. as abscissas, for steam expanding adiabatically from an initial pressure of 400 lb. per sq. in. abs. and a temperature of 750 deg. F. to a back pressure of 1.5 in. of mercury abs. If the total energy available were used in six pressure stages of a steam turbine, find the pressure drop for each stage.

12. An impulse steam turbine received 1000 lb. of dry steam per hour at a pressure of 225 lb. per sq. in. abs. and exhausted into a 27-in. vacuum. Quality, 70 per cent at exhaust. Barometer, 28.5 in. mercury. The rotor was 36 in. in diameter and operated at 2000 r.p.m. (*a*) Find the number of velocity stages. (*b*) How many pressure stages would be required? (*c*) Make a sketch of the arrangement of nozzles, rotors, stators, etc., for *a* and *b*; also, a pressure-velocity graph for each showing the variation in pressure and velocity as the steam passed through the turbine.

13. Determine the number of *pure* reaction stages of a steam turbine operated at 3000 r.p.m. when supplied with 5400 lb. of steam per hour at a pressure of 138 lb. per sq. in. abs. and a temperature of 440 deg. F. The steam was expanded to a back pressure of 3 in. of mercury abs.

14. If the mean diameter of the rotor of the steam turbine in problem 13 was 30 in., calculate the number of *impure* reaction stages needed.

15. A steam turbine is to operate at a speed of 3600 r.p.m. and the pitch diameter of the rotor blades is to be 42 in. The probable steam consumption will be 30,000 lb. per hr. with the supply steam at 200 lb. per sq. in. abs. and 150 deg. F. superheat, and the condenser pressure 2 in. of mercury abs. Assuming adiabatic expansion and neglecting friction losses, determine the number of stages required in each of the following cases: (*a*) all pressure (Rateau) stages; (*b*) first stage two-row velocity (Curtis), others Rateau; and (*c*) all *impure* reaction stages.

16. A steam turbine has two velocity stages. The steam is directed against the first row of blades with a velocity of 4000 ft. per sec., at the rate of 5400 lb. per hr. The steam is discharged from this row at a velocity of 2100 ft. per sec. and is then directed upon the second row of blades from which it is discharged at a velocity of 150 ft. per sec. (*a*) Neglecting friction and other losses, how many horsepower will this turbine develop? (*b*) How much power would be developed if frictional effects reduced each of the velocities 15 per cent? (*c*) Find the steam consumption in pounds per horsepower-hour for part *b*.

17. The steam supply pressure to a single-stage impulse turbine is 135 lb. per sq. in. abs. and the exhaust pressure 14 lb. per sq. in. abs., the steam being initially of 98 per cent quality. The peripheral speed of the blades is 2000 ft. per sec. and the nozzle angle 20 deg. (*a*) Draw the velocity diagram determining the blade angle and the absolute exit velocity, neglecting friction and assuming adiabatic flow. (*b*) Find the actual and theoretical work done on the blades per second, per pound of steam. (*c*) Determine the efficiency of the turbine.

18. Draw the theoretical velocity diagram for a 750-kw., two-stage Curtis turbine

operating under the following conditions: Initial pressure, 175 lb. per sq. in. abs.; superheat, 150 deg. F.; and back pressure, 2 in. of mercury abs. Nozzle angle 20 deg. and peripheral velocity 500 ft. per sec. There are two rotating elements and one stationary element in this turbine. Assume equal heat drop in each stage, and neglect residual energy.

19. The steam supply pressure to a single-stage impulse turbine was 160 lb. per sq. in. abs. and the exhaust pressure 3 in. of mercury abs., the steam being initially dry and saturated. The turbine was connected to an electric generator through reduction gears. (*a*) Find the thermal efficiency of the unit, based on 35 lb. of steam used per hour per kilowatt delivered at the switchboard. (*b*) Assuming that 25 kw. are delivered by the generator and that 5 kw. are required to overcome friction in the turbine, gears, and generator, determine the quality of the exhaust steam, neglecting radiation losses. (NOTE.—Work is done in the turbine by part of the heat in the steam.)

20. A turbine-generator delivered 750 kw. and required 25 lb. of dry steam per kilowatt-hour; steam pressure, 180 lb. per sq. in. abs.; and exhaust, 4 in. of mercury abs. The boiler supplying steam to the above turbine received feedwater at 196 deg. F., and the evaporation from and at 212 deg. F. was 8 lb. of water per pound of coal as received. Coal as received contained 12,200 B.t.u. per lb. What was the heat efficiency of the plant?

21. Using Fig. 217, calculate the unit water rate of the turbine when delivering 800 kw. and bleeding 5000 lb. of steam per hour. Also find the total heat per pound of steam extracted.

22. Determine the weight of water stored in a steam accumulator to operate a 1000-kw. low-pressure steam turbine for 5 minutes. The steam supply was entirely cut off during this time and the pressure dropped from 15 to 12 lb. per sq. in. abs. The water rate of the turbine was 30 lb. per kw-hr.

23. A compound turbine received steam at 700 lb. per sq. in. abs. and 850 deg. F. After expansion to the saturated state in the high-pressure cylinder the steam was resuperheated to 400 deg. F. and then expanded in the low-pressure cylinder to an exhaust pressure of one pound absolute. Assuming adiabatic expansion: (*a*) Find the exhaust pressure of the high-pressure cylinder. (*b*) What was the quality of the steam at the low-pressure cylinder exhaust? (*c*) What percentage of the total power of the turbine was developed in the high- and in the low-pressure cylinders? (*d*) Find the reheating cycle efficiency. (*e*) Determine the Rankine cycle efficiency between the original and final steam conditions.

24. In a plant operating on the ideal regenerative cycle, steam entered the turbine at 400 lb. per sq. in. abs. and 600 deg. F.; exhaust was at 2 in. of mercury abs. (*a*) Find the Rankine cycle efficiency. (*b*) Find the regenerative cycle thermal efficiency.

CHAPTER XVI

STEAM-CONDENSING EQUIPMENT

258. Condensing Operation.—Theoretically, the lower the back pressure of a steam engine or turbine the greater will be the amount of work that can be obtained from a given weight of steam. By decreasing the exhaust pressure with unchanged cylinder volume, the area W_1 in Fig. 223 represents the work gained, whereas using the steam in a turbine corresponds to a continuation of the expansion line as the exhaust pressure decreases, and the further gain is represented by the work area W_2. It is desirable therefore to exhaust into a space in which the lowest possible pressure exists, when the work obtained per pound of steam is the only consideration.

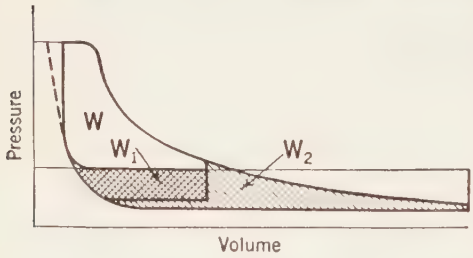


FIG. 223.—Diagram showing gain in work resulting from condensing operation.

A condenser is used in connection with steam engines and turbines for the purpose of obtaining one of two important results, and generally both of them: (1) a reduction of back pressure with the consequent increase in m.e.p. of the prime mover, and (2) the recovery of the condensed steam, which may be returned to the boiler as practically pure water.

The condensation of steam in a closed vessel produces a partial vacuum by reason of the great reduction in volume of the vapor. A pound of dry steam at 14.7 lb. per sq. in. abs. pressure has a volume of 26.82 cu. ft. Theoretically, if this quantity were contained in a steam-tight vessel of 26.82 cu. ft. volume at 14.7 lb. per sq. in. abs. pressure, and if condensation into water occurred at 142 deg. F., the liquid would occupy only 0.0163 cu. ft. The volume of the liquid would be $\frac{1}{1615}$ of the space inside of the vessel, and the absolute pressure would fall to 3 lb. per sq. in. abs. These figures would be modified in actual practice by the pressure of air and other non-condensable gases within the condenser, and the pressure in the vessel would be considerably higher.

Steam condensers are devices in which the condensation of the

exhaust steam, and the removal of air and other non-condensable gases, are continuous processes. A certain amount of air is present in water obtained from natural sources. Air may also leak into the condenser through the shell, the piping, the turbine casing, and the various packings. The degree of *vacuum* (Art. 7) which can be obtained with condensing apparatus depends largely upon the tightness of the joints and the amount of air which enters with the steam, as well as the temperature to which the condensed steam (condensate) can be reduced. When the pressure is reduced to below atmospheric, the entrained air leaves the condensed steam in the surface condenser; air also leaves both the steam and the cooling water in the jet and barometric types of condensers. In any case, the air remains in the condenser until positively removed.

Experience has shown that the highest practicable vacuum suitable for ordinary reciprocating steam engines is 26 in. of mercury referred to a 30-in. barometer. Higher vacua are not economical. In the reciprocating engine, full advantage of higher vacua cannot be obtained by expanding steam to the condenser pressure, on account of the excessively large volumes of steam produced. Attempts to do so largely increase the bulk of a single cylinder or diminish the mechanical power of the engine, owing to the early cut-off required; also, the low-temperature exhaust steam would cool the cylinder walls to cause a very rapid increase in cylinder condensation.

For steam turbines the average practice is to employ a vacuum of 28 in. of mercury referred to a 30-in. barometer. Many installations under 1000 kw. operate on 26 to 27 in., and others of larger capacity exhaust to $28\frac{1}{2}$ to 29 in. of vacuum referred to a 30-in. barometer. There is a tendency on the part of steam-turbine manufacturers, because of the obvious advantages of steam turbines over reciprocating engines for operation at high vacua, to call attention to the reduction in steam consumption when a plant employs a high vacuum. In obtaining this economy, consideration must be given to the greater first cost of the condenser, pumps, and piping, together with the increased operating expense. Local conditions, particularly the amount and temperature of the cooling water, determine in part the feasibility of a condenser where condensing operation is desirable. In locations which are practically at sea level and where there is an adequate supply of low-temperature cooling water, it would probably be profitable to install condensing apparatus of sufficient size to operate steam turbines at $28\frac{1}{2}$ to 29 in. of vacuum referred to a 30-in. barometer.

259. Classification of Condensers.—Condensers used in steam plants may be divided into two general classes: (1) *jet condensers*, in which condensation of the exhaust steam is effected by direct contact of the cooling

water and the vapor; and (2) *surface condensers*, wherein the exhaust steam and the cooling medium do not come in direct contact with each other, the heat being extracted from the steam by heat transfer through metal walls.

More surface condensers are built today than any other type. The surface condenser is used exclusively for marine purposes, as the recovery of the condensate is essential. For large steam-turbine installations, the use of jet condensers is exceptional, as, apart from the recovery of the condensate, the auxiliary power consumption of a jet-condenser plant, and the first cost of the requisite air pumps to attain the extremely high vacua desired, represent serious drawbacks. However, with moderate-size turbine units and for reciprocating steam engines, the jet type has a large field of useful application, especially where an abundant supply of good feedwater is available.

Surface condensers generally consist of a cast-iron or mild-steel shell, fitted with a tube plate at each end. A great number of tubes extend between the tube plates, thus forming the cooling surface. In most cases, steam is admitted to the condenser through an opening in the top of the shell, and water is passed through the tubes. Where the condenser is used in conjunction with a pumping engine, the reverse arrangement is generally adopted, that is, the water passes over the outside of the tubes and the steam is allowed to pass through the tubes. The latter arrangement is called a *water-works condenser*. Another form of surface condenser, known as an *evaporative condenser*, has the shell omitted. The steam is fed to the condenser tubes over whose outer surface water is sprayed. The cooling is effected mainly by evaporation of the water in the surrounding atmosphere.

Jet condensers may be classified as low-level jet and barometric condensers. The two types are similar in the manner in which the exhaust steam and cooling water are brought together, but differ in the method of removing the water and the condensate. In the low-level jet condenser the condensing chamber is at a low elevation, water being removed by a pump, and the overall height of the unit is low enough so that the condenser may be directly beneath the turbine or engine. In the barometric type, the condenser is placed at a sufficiently high level (about 40 ft.) above the point of water discharge to enable the water to drain away by gravity through a water-sealed " tail pipe " or barometric column.

260. Low-level Jet Condensers.—Figure 224 shows a low-level jet condenser of the parallel-flow design, that is, the steam, cooling water, and non-condensable gases flow in the same direction. The inlet cooling water enters an annular chamber at the top and is sprayed into the condenser chamber through a large number of helical distributing noz-

zles. These nozzles break up the water into a finely divided state, thus insuring intimate contact with the incoming steam. The mixture of cooling water and condensate passes through a cone in which the air is entrapped in the water as far as possible. The mixture then falls in a solid stream through the lower part of the condenser body into the pump opening. The water is lifted from the source of supply to the nozzle injection head by means of the vacuum within the condenser, and removed by a centrifugal pump working against the vacuum in the condenser. The surplus air-vapor mixture at the lowest temperature is withdrawn from near the top of the condenser body into the suction inlet of an air pump or air ejector. In order to prevent the water from rising to an excessively high level in the condenser, a vacuum breaker is provided. This breaker consists of a float-controlled valve of generous size. When the water level reaches a predetermined height, the valve opens and admits air, diminishing the vacuum so that less water is drawn into the condenser. The vacuum breaker is not in action during normal operation, but is provided as a safety device in case the motor driving the pumps should stop, thus failing to remove the accumulated water, or in case the water injection valve is improperly adjusted. The removal pump shown is of the vertically split casing type which permits quick disassembly of the unit.

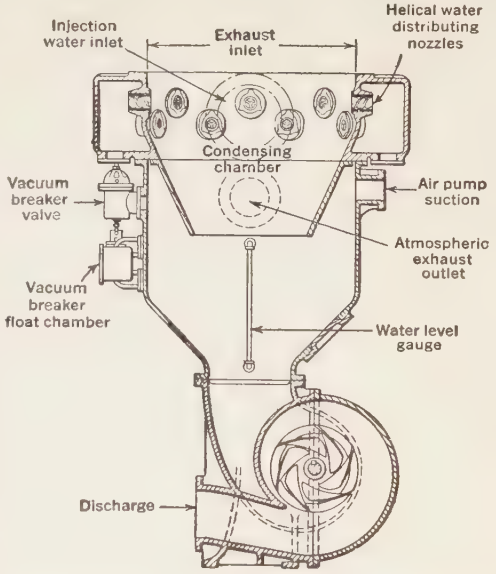


FIG. 224.—Foster-Wheeler low-level jet condenser.

Jet condensers in which the cooling water, condensate, and non-condensable gases are withdrawn by a single pump are called low-level, low-vacuum jet condensers because of the limited air capacity of the pump. In the condenser illustrated by Fig. 224, the non-condensable gases are removed by a separate pump or ejector, higher vacua are obtainable, and the condenser is known as a low-level high-vacuum jet condenser. Under certain conditions the air and water can be removed by the kinetic action of the steam, in which case the jet condenser is designated as an *ejector* or *siphon condenser*.

The most widely adopted type of ejector condenser is the low-level *multijet condenser*, illustrated by Fig. 225. The condenser consists of a closed cylindrical chamber, in the upper end of which is a water-nozzle case. The latter is fitted with a Venturi-shaped combining tube, made up of several sets of tapering rings joined together by equally spaced ribs. The combining tube is connected with a tail pipe, the lower end of which is immersed in water in a well or reservoir. The water-nozzle case is provided with a nozzle plate containing the water nozzles.

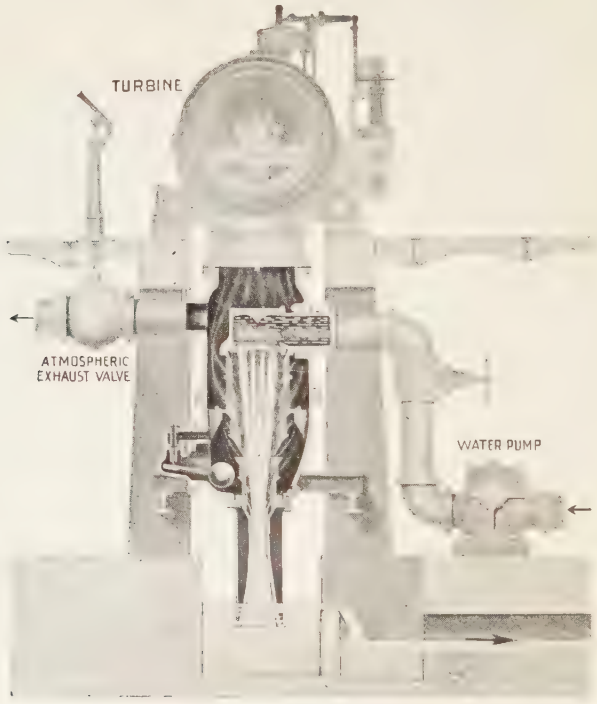


Fig. 225.—Koerting low-level multi-jet ejector condenser under a steam turbine.

The injection water, Fig. 225, is forced under pressure into the water-nozzle case of the condenser and passed through the water nozzles, which are designed to discharge a specified amount of water according to the conditions of flow determined by the water pressure and the existing vacuum. As the water jets emerge from the nozzles they are directed into the throat of the combining tube, where they unite to form a single jet. The exhaust steam enters the condenser chamber through the steam inlet, flows through the annular passages between the tapering rings, comes into direct contact with the converging water

jets, and is condensed. From the combined effect of the external water pressure, the vacuum existing in the condenser, and gravity, the multiple water jets attain a velocity sufficient to entrain the condensed steam, air, and non-condensable gases, and to discharge them into the hot well against the pressure of the atmosphere. The water jets create a vacuum by condensing the steam and maintain the vacuum by entraining and removing the air and non-condensable gases. No air pump or water removal pump is required.

A float-operated vacuum breaker is provided to admit air to the vacuum chamber when the flow of injection water fails. At that time, the vacuum is displaced by water from the hot well through the diffuser tube. The breaker protects the turbine or engine exhaust lines against the danger of becoming flooded with water. An atmospheric exhaust valve is connected to the top of the condenser, as shown in Fig. 225. This valve opens automatically when the pressure in the condenser becomes greater than that of the atmosphere, and closes when there is a partial vacuum in the condenser. When the atmospheric exhaust valve is open, the turbine exhaust steam flows to the atmosphere. The connection shown between the turbine and the condenser consists of a corrugated copper tube with cast-iron flanges. This joint permits movement due to expansion and contraction because of temperature changes.

The ejector condenser requires a larger amount of water than other forms of jet condensers because it is not practicable to subdivide the water jets as effectively, but the absence of vacuum pumps may offset this disadvantage. Multijet condensers can maintain vacua up to 29 in., referred to a 30-in. barometer, with 60 deg. F. cooling water; this type of condenser is suitable for turbine installations up to 10,000-kw. capacity.

261. Barometric Condensers.—This type of jet condenser consists essentially of a head, similar to that used with the low-level jet condenser, and a tail pipe or barometric column which partly or wholly takes the place of the removal pump. Figure 226 shows a section through a barometric condenser of the counter-flow type. In the condenser illustrated, the water enters a pool in the upper part of the condensing head at a point above the steam inlet. Ample opportunity is given for the air content of the water to be removed by the vacuum pump. The water partially relieved of the air then meets the entering steam, and absorbs the latent heat. The air released from the steam passes upward through the falling sheets of water to a point above the water pool, and is removed by the vacuum pump.

The hot water resulting from the mixing process falls to the bottom of the condenser, and the air at the extreme top is cooled to practically

the temperature of the inlet water. This cooling of the air causes condensation of the water vapor accompanying it, and the condensed vapor collects in the space surrounding the air cooler and flows down into the drain pipe. The vacuum pump, usually called a *dry-air pump* or *dry-vacuum pump*, handles cool gases containing little vapor and practically no water. The hot water flows downward into the tail pipe, which is about 35 ft. in length. The lower end of the tail pipe dips into the hot well. As atmospheric pressure can support a column of water about 34 ft. high, theoretically, the tail pipe forms an automatic removal pump, water flowing from the tail pipe as rapidly as it accumulates.

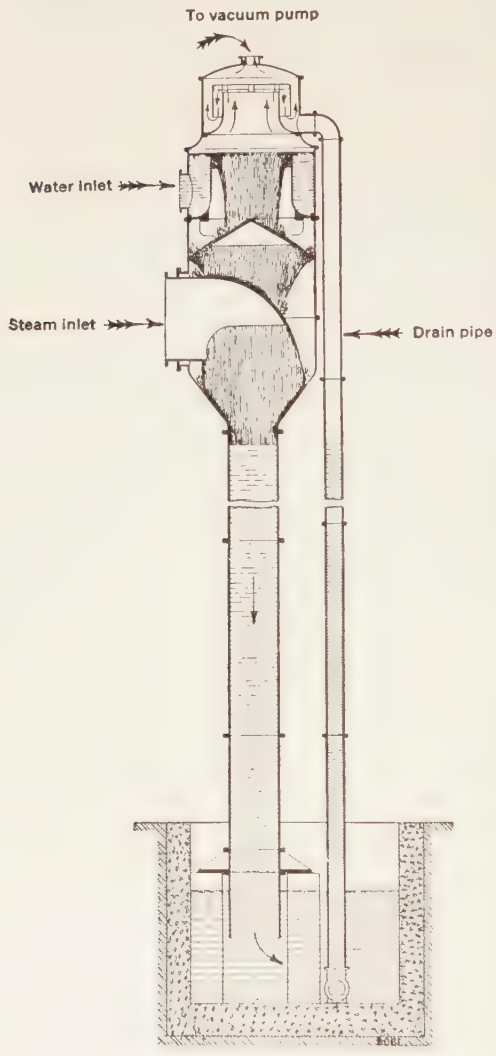


Fig. 226.—Ingersoll-Rand barometric condenser.

In both low-level and barometric condensers the custom is to lift the water from the source of supply to the injection head by means of the vacuum within the condenser. The greatest height to which the water can be lifted by this means is about 18 ft. with a 28-in. vacuum, referred to a 30-in. barometer. Where necessary, a pump of suitable type is used to aid in lifting the water to the condenser head. The construction of the barometric condenser is simple, there being no moving parts, no nozzles and contracted orifices to become clogged, and no need for vacuum breakers. Barometric condensers require a long exhaust pipe because of the long tail pipe. The following disadvantages result: (1) greater cost of piping, (2) the

possibility of air leakage, and (3) the increased resistance offered to the flow of exhaust steam.

262. Surface Condensers.—The surface condenser does not permit the cooling medium and the exhaust steam to come into contact with each other. The device consists essentially of a cylindrical vessel through which pass numerous brass or metal alloy tubes, Figs. 227 and 228. In the usual surface condenser, the cooling water circulates through the tubes, and the steam to be condensed is brought in contact with the outer surfaces of the tubes. Condensers of this type may be "single-pass," in which the water flows in one direction only through all the tubes, or "two-pass," in which the water flows in one direction through part of the tubes and then returns through the remainder of the tubes. Surface condensers may operate on the wet vacuum or the wet and dry vacuum principle. In the case of wet vacuum operation, Fig. 227, one pump (wet-air pump) connected to the bottom of the condenser chamber removes the condensate and along with it the large volume of air and non-condensable gases. With the wet and dry vacuum system, Figs. 228, 229, and 230, the condensate is removed from the lowest point by one pump and the air at some point higher up in the condenser by a dry-air pump or ejector.

A surface condenser and its auxiliaries should accomplish the following results: (1) *Steam* should enter the condenser with the least possible resistance, and the pressure drop through the condenser should be reduced to a minimum. (2) *Air*, a poor conductor of heat, should be rapidly removed from the heat-transmitting surfaces. The air should be collected at suitable places, practically free from entrained water vapor and cooled to a low temperature. The air removal should be with a minimum volume and the least expenditure of mechanical energy. (3) *Condensate* should also be rapidly cleared from the heat-transmitting surfaces, freed from air, collected at suitable points for removal, and returned to the boiler at the maximum practicable temperature. (4) *Circulating water* should pass through the condenser with little friction, with a minimum amount of deposited materials, and with a maximum amount of heat absorption.

An elementary form of surface condenser, mounted over a combined circulating water and wet-air pump of the piston type, is shown by Fig. 227. The exhaust steam enters at the top of the shell and passes downward over the water-cooled tubes. A baffle plate is located under the exhaust inlet in order to distribute the steam properly over the rows of tubes and to prevent erosion of the upper row. The condenser tubes are usually made of ordinary brass (62 copper, 38 zinc), or Muntz metal (60 copper, 40 zinc) for pure water, and Admiralty brass (70 copper, 29 zinc, 1 tin) for salt water and impure river waters.

The outside diameter of the tubes varies from $\frac{5}{8}$ to 1 in. The condensate mixed with gases and vapors is removed from the bottom of the condenser by the wet-vacuum pump and discharged to the hot well. The cooling water is forced through the tubes of the condenser by means of the reciprocating circulating-pump. The water enters the lower tubes at the right end, makes two "passes" through the condenser, and leaves at the top. The vertical support plate at the center of the condenser is usually made of cast iron and in long-tube types is so spaced as to allow not more than 6 ft. of unsupported tube. This type, Fig. 227, of condenser equipment is intended for steam engines and

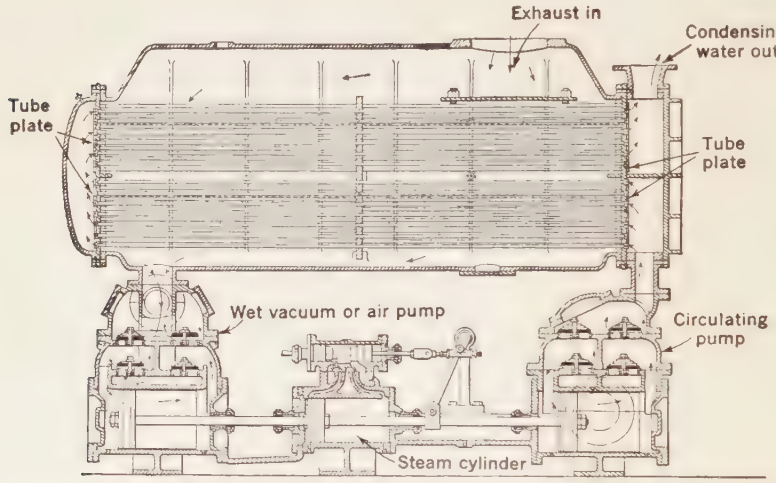


FIG. 227.—Elementary surface condenser and equipment, wet-vacuum operation.

small turbines where the vacuum desired does not exceed 26 in. of mercury referred to a 30-in. barometer.

With the arrangement illustrated by Fig. 227, the steam which condenses upon the upper tubes falls as rain from tube to tube until it settles at the bottom. The outer surfaces of the lower tubes are, therefore, practically covered with water, which results in two disadvantages. First, these tubes carry the coolest circulating water and cool the condensate coming in contact with them while the water flowing through the tubes is unnecessarily heated. Cooling of the condensate gives a lower hot-well temperature than would otherwise be obtained. If the condensate is to be used for boiler feedwater, the temperature in the hot well should be maintained as high as possible, since this water must again be heated to boiler temperature with a corresponding expenditure of heat. Second, tubes which are being used to cool water

covering them are of little value as condensing surface, so that such surface in a condenser is practically useless.

A condenser which embodies the most modern features of design and arrangement is shown by Figs. 228 and 229. Each tube bank presents a maximum number of front line tubes to the incoming steam, and also provides exit lanes to the air-cooler section. Steam passes from the inlet dome through either of the tube layers toward the exit lanes and thence to the air cooler. The shallow tube layers combined with the unimpeded flow of air and vapor through the exit lanes results in a low pressure drop through the condenser. The presence of air around a

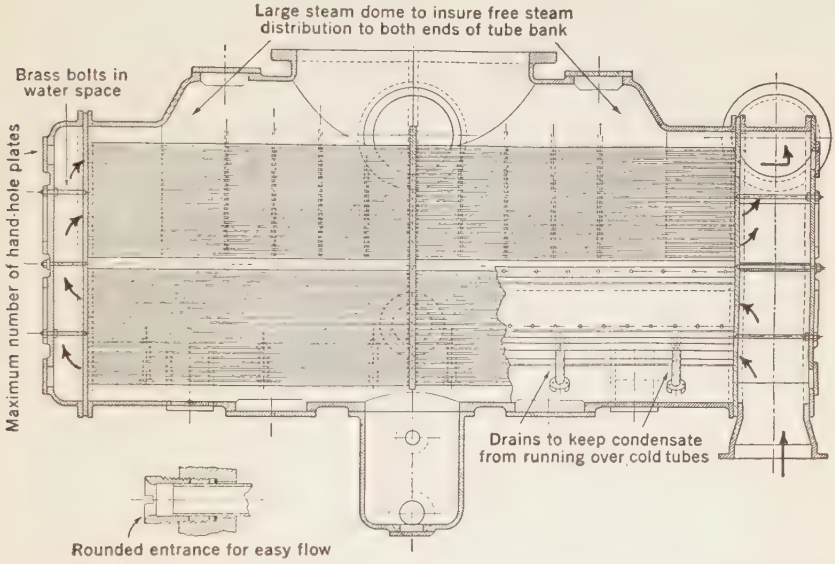

Large steam dome to insure free steam distribution to both ends of tube bank
Brass bolts in water space
Maximum number of hand-hole plates
Drains to keep condensate from running over cold tubes
Rounded entrance for easy flow

FIG. 228.—Longitudinal section of Worthington surface condenser, 15,000 to 25,000 sq. ft. of cooling surface. Wet- and dry-vacuum operation.

condenser tube is detrimental to the transfer of heat through the tube; with the condenser design of Figs. 228 and 229, the air comes in contact with the least possible number of tubes. This feature, combined with the effective distribution of steam to the tubes, insures a high rate of heat transfer and a high vacuum.

The air-cooler section, Figs. 228 and 229, consists of a group of tubes insulated from the steam space by a double thermos-type baffle and supplied with the coldest circulating water. The air cooler lowers the temperature of the air and vapors, thus reducing the volume of the mixture to be handled by the air pump. A hot well is attached to the lowest portion of the condenser shell; see Fig. 228. In the double-tube-layer design, a central steam lane between the tube banks feeds steam

to the lower tubes and conducts steam to the hot well; thus the temperature of the condensate is raised to approach that of the exhaust steam.

An installation of a surface condenser with the necessary auxiliaries as applied to a steam turbine of moderate size is shown by Fig. 230. The condensate is removed by a centrifugal hot-well pump. The cooling water is supplied by a circulating (centrifugal) pump. The non-condensable gases are removed near the bottom of the condenser by means of a steam air ejector. The exhaust steam from the turbine may be bypassed to the atmosphere through the exhaust relief valve, which opens automatically when the pressure in the condenser becomes greater than that of the atmosphere. The condenser body is flexibly connected to the turbine by means of a corrugated copper or a heavy rubber expansion joint. In addition to an expansion joint, or when no expansion joint is provided, between the turbine and condenser, the condenser may be mounted on spring supports to take care of expansion and contraction.

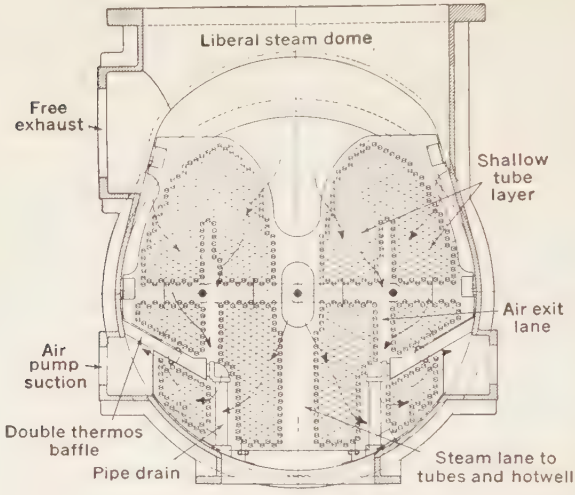


FIG. 229.—End section of Worthington surface condenser, showing double-folded tube layer design.

263. Vacuum Pumps.—The vacuum or air pumps used in connection with steam-condensing plants are of four distinctive types: (1) reciprocating-piston, (2) positive rotary-displacement, (3) hydro-centrifugal, and (4) steam-jet. The earliest dry-vacuum pumps were merely air compressors of the piston type which worked between a high vacuum and atmospheric pressure. These pumps receive saturated "air" at the pressure existing in the condenser and compress it to a pressure high enough to discharge it to the atmosphere. Some of these reciprocating pumps are arranged with a crank and flywheel and are known as rotative dry-vacuum pumps. Leakage, defective jacket cooling, and re-expansion of the air in the clearance spaces make it difficult to maintain a vacuum of more than 26 in. of mercury referred to a 30-in. barom-

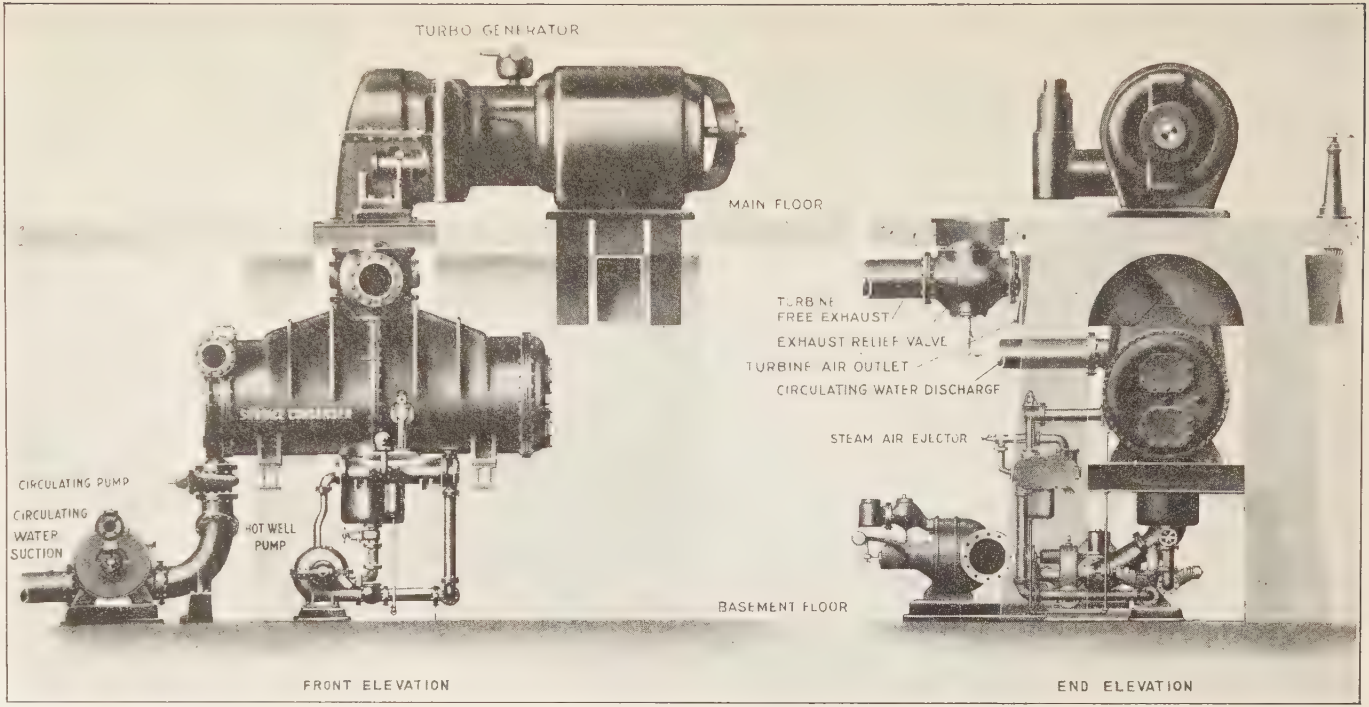


FIG. 230.—Typical surface condenser layout for steam turbine with turbine-driven circulating pump, steam air-ejector as air removal apparatus, and motor-driven hot-well pump.

eter. Another disadvantage of reciprocating air pumps is the excessive size and high cost when made in large capacity. Positive rotary displacement vacuum pumps are similar in construction to the blower shown by Fig. 251, in which air from the condenser is drawn through the intake and discharged to the atmosphere.

An example of a hydraulic air pump, consisting primarily of a multivane wheel in conjunction with an ejector, is shown by Fig. 231. Water taken from a circulating tank enters the central chamber of the pump, from which it is hurled by centrifugal force through the diffuser N. The water is caught up by the blades V of the wheel, rotating at a suitable speed. The blades divide the water into a succession of small elements which are impelled or hurled down the diffuser pipe N' in the form of thin sheets. These sheets form a series of water pistons; the air entering at the top is trapped between successive sheets and carried out against the atmospheric pressure.

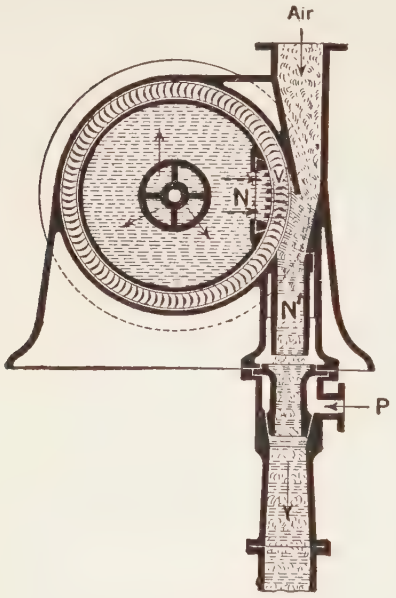


Fig. 231.—Westinghouse-Leblanc hydraulic air pump.

The hydraulic air-pump is a positive displacement pump in which water pistons are used. In starting, water or steam under the necessary pressure can be admitted at P, thus creating sufficient vacuum to establish the flow of water and air. The water is used over and over again with an addition of about 10 per cent make-up to keep down the temperature, since very little heat is abstracted from the air-vapor mixture. In some installations, the hurling water is recirculated through cooling coils so that discharge to overflow and make-up for lowering the temperature are not necessary. The hydraulic type of air pump is used with larger condenser installations than the reciprocating-piston type chiefly because of its compactness, high air-removal capacity, ability to maintain a slightly higher vacuum, and ability to carry overloads. However, the hydraulic air-pump is slower in starting and requires more power than the reciprocating pump.

The steam-air ejector vacuum pump has practically supplanted other types for steam condensers where high vacua are required and where large volumes of gases must be handled, because of its compact-

ness, absence of moving parts, simplicity of operation, and high efficiency. One or more steam jets moving at a high velocity (about 3000 ft. per sec.) entrain the saturated non-condensable gases, and the mixture passes through an expanding tube in which the velocity is reduced with a corresponding increase in pressure. The capacity varies with the steam pressure and the size and number of steam nozzles.

Typical American practice is illustrated by Fig. 232, a two-stage ejector with a combined inter- and after-condenser of the surface type. In this equipment the steam, with entrained air and vapor from the first-stage tube, passes into the inter-condenser where it is condensed,

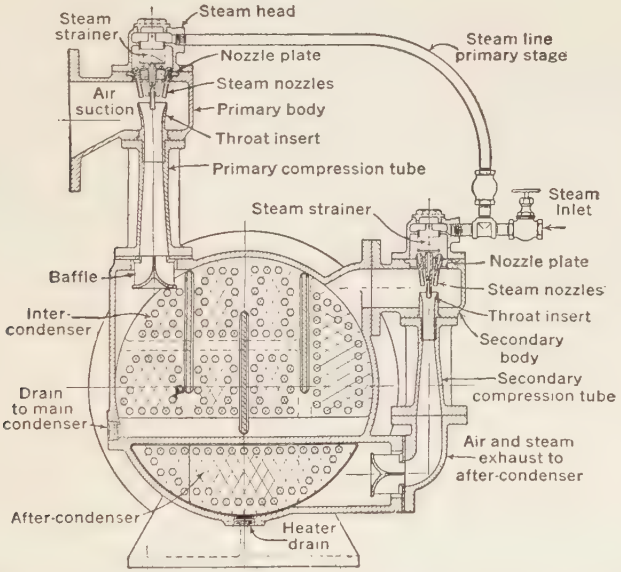


Fig. 232.—Section through Foster-Wheeler two-stage steam-jet air pump with surface inter- and after-condenser.

the condensate falling into a receiving well below. From this point the inter-condenser condensate is drained through a U-tube seal into the main condenser hot well. The non-condensable gases of the inter-condenser are drawn off through the second-stage tube of the ejector pump and discharged into the after-condenser at a pressure slightly above that of the atmosphere. The condensate from the main condenser may be used for cooling the ejector condensers. The condensate from the after-condenser is generally drained to an open feedwater heater; the temperature of the drains from the after-condenser is about 200 deg. F., whereas the temperature of the inter-condenser drains averages 130 to 150 deg. F.

264. Heat Supplied to the Condenser.—The size of the condensing equipment is determined by the required heat exchange—i.e., the amount of heat to be transferred from the steam to the cooling water. This heat transfer depends upon the quantity and heat per pound of the steam, and the quantity and temperature of the cooling water. Where the condenser is bolted directly to a simple non-reheating and non-extraction [1] steam turbine or engine-generator set, the *heat supplied to the condenser* is that supplied to the prime mover, exhausting to the condenser, less that converted into work and consumed by generator losses, friction, radiation, and other losses not readily determined. Hence

$$h_x = h_t - \frac{3413K}{W_s \times e_{me}}; \tag{134}$$

in which h_x = total heat of steam at exhaust from prime mover, B.t.u. per lb. above 32 deg. F.

h_t = total heat of steam at throttle to prime mover, B.t.u. per lb. above 32 deg. F.

3413 = heat equivalent of one kilowatt-hour, B.t.u. per kw. per hr.

K = load on prime mover, kw.

W_s = total steam supplied to throttle of prime mover, lb. per hr.

e_{me} = combined mechanical and electrical efficiency of prime mover complete, expressed decimally, including all friction, electrical loss, windage external to the turbine, radiation, etc.

When the load K on the prime mover is given in horsepower the constant 2545 must be substituted for 3413. The value of e_{me} varies with both the type and size of prime mover, and may be assumed from general practice or obtained from the manufacturer.

265. Injection Water Required for Jet Condensers.—In jet condensers the cooling water and condensate are discharged as a mixture and therefore have the same average discharge temperature. Neglecting the effect of radiation, leakage, and entrained air, the heat given up by the exhaust steam is equal to the heat received by the injection water.

$$W_s(h_x - h_{f.2}) = W_w(t_2 - t_1),$$

[1] Extended equations for h_x when considering steam reheating and extraction prime movers are given in Section 14, "Test Code for Condensing Apparatus," The American Society of Mechanical Engineers, November, 1927.

or

$$W_w = \frac{W_s(h_x - h_{f,2})}{(t_2 - t_1)}; \tag{135}$$

in which W_w = weight of injection water required, lb. per hr.
$h_{f,2}$ = heat of the liquid of condensate leaving the condenser, B.t.u. per lb. above 32 deg. F.
t_2 = temperature of mixed cooling water and condensate leaving condenser, deg. F.
t_1 = temperature of injection water entering condenser, deg. F.

Other items as in equation 134.

The value of t_2 would be the same as that corresponding to the absolute pressure in the condenser if it were not for the pressure exerted by the non-condensable gases. In high-vacuum jet condensers the air pressure is low and the temperature t_2 ranges from 6 to 10 deg. F. below that of the steam temperature corresponding to the condenser pressure. In the usual low-vacuum jet condenser, t_2 may be 15 to 25 deg. F. less than the steam temperature. The weight of injection water required per pound of steam will vary from 20 to 60 lb., depending mainly upon the initial temperature of the water and the vacuum desired. Experience has shown that 10 per cent or more should be added to the calculated values of W_w from equation 135 to obtain the weight of injection water which would probably be used.

266. Circulating Water Required for Surface Condensers.—Attempts have been made to dispose of the heat rejected in a surface condenser by passing air over the cooling surfaces. The power consumption involved in attaining the necessary cooling effect rules the air-cooling method out of practical consideration except in locomotive practice. The only effective cooling medium is water. The weight of cooling water varies with the design of the condenser and the conditions of operation, such as diameter, thickness and spacing of the tubes, cleanliness of tube surfaces, initial temperature and velocity of the circulating water, vacuum desired, etc. The quantity of circulating water required can be determined approximately by calculation, and the values thus found must then be increased by such factors as experience has shown to be necessary.

For surface condensers, the heat given up by the exhaust steam equals the heat absorbed by the circulating water, neglecting the effect of radiation, leakage, and entrained air:

$$W_s(h_x - h_{f,c}) = W_w(t_2 - t_1),$$

or

$$W_w = \frac{W_s(h_x - h_{f.c})}{(t_2 - t_1)}; \tag{136}$$

in which W_w = weight of circulating water required, lb. per hr.
$h_{f.c}$ = heat of the liquid of condensate leaving the condenser, B.t.u. per lb. above 32 deg. F.
t_2 = temperature of circulating water leaving condenser, deg. F.
t_1 = temperature of circulating water entering condenser, deg. F.
Other items as in equation 134.

The discharge temperature t_2 of the circulating water is generally 10 to 20 deg. less than the temperature corresponding to the steam pressure in the condenser. The temperature of the condensate is often 15 or more degrees below that of the exhaust steam, but good design allows the condensate temperature to approximate very closely the temperature corresponding to the total pressure in the condenser. The weight of circulating water required per pound of steam will vary from 25 to 50 lb. with low vacua and 60 to 120 lb. for vacua over 27 in. of mercury, referred to a 30-in. barometer. For steam-engine condensers, one square foot of cooling surface is generally allowed for every 12 lb. of steam condensed per hour from 24 to 26 in. vacuum, referred to a 30-in. barometer, with 70 deg. F. circulating water. An average figure for steam-turbine condensers using high vacua is 8 lb. of steam condensed per hour per square foot of tube surface. In some of the recent central-station installations, surface condensers of the vertical single-pass type are employed; the surface used varies from 0.65 to 1.15 sq. ft. per kw. of turbine rating. For the horizontal single- and two-pass types, about 0.90 to 1.75 sq. ft. per kw. are required. Small turbine-generators require from 2 to 4 sq. ft. of tube surface per kilowatt.

267. Heat Transfer in Surface Condensers. The performance of a surface condenser is expressed in B.t.u. transmitted per square foot of tube surface per hour per degree mean difference in temperature between the two sides of the tube. Neglecting radiation and leakage, the heat absorbed by the circulating water is equal to the heat given up by the exhaust steam, or

$$UAD_m = W_s(h_x - h_{f.c}),$$

and

$$A = \frac{W_s(h_x - h_{f.c})}{U \times D_m}; \tag{137}$$

in which A = area of cooling surface, on steam side, sq. ft.

U = mean coefficient of heat transmission, B.t.u. per hr., per sq. ft. of tube surface, per deg. F. mean difference in temperature.

D_m = mean temperature difference between the steam and circulating water, deg. F.; see equations 138 and 140.

Other items as in equations 134 and 136.

The coefficient U, as used in equation 137, refers to the *mean* or average value for the entire tube surface, and not the *actual* value, since

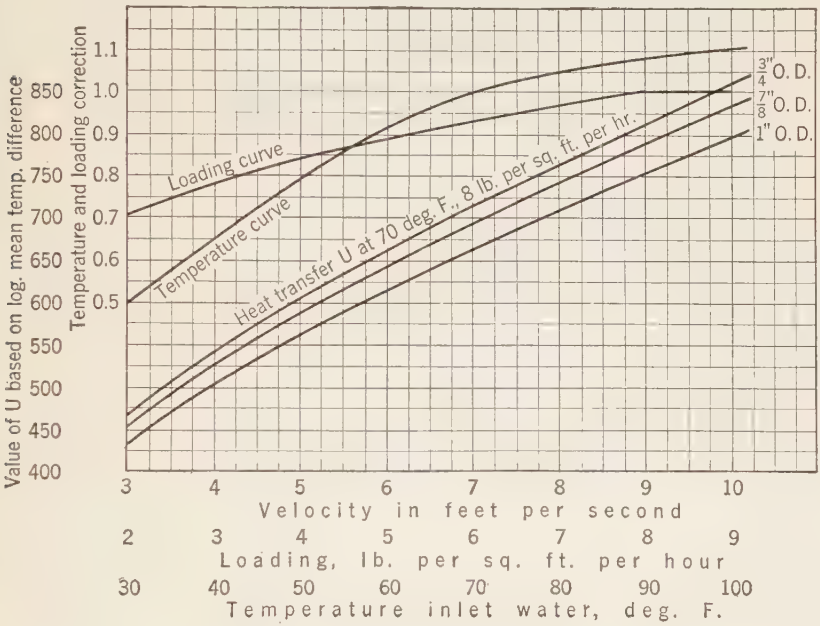


Fig. 233.—Heat transfer curves for surface condensers.

the latter varies widely for different parts of the condenser. The actual value varies from more than 1000 B.t.u. for air-free steam in the upper part of the condenser to about 5 B.t.u. for tubes surrounded only by air (as approximated in the lower part of a surface condenser). The results of tests on various makes of surface condensers formed the basis for the coefficients of heat transfer indicated by Fig. 233. These curves show that the coefficient increases with an increase in water velocity through the tubes; that it varies with the inlet water temperature, being greater with high than with low water temperatures; and that tubes of small diameter give a somewhat higher coefficient of transfer than those of larger diameter. It is also known that the condition of

both tube surfaces with respect to cleanliness, pitting, and scale largely influences the heat transmission, the greatest transfer taking place when the tube surfaces are perfectly clean and smooth. In any case, high vacua in condensers are primarily dependent upon the lowest possible inlet cooling water temperature.

Figure 233 shows heat-transfer curves for three tube sizes, viz., $\frac{3}{4}$-in., $\frac{7}{8}$-in., and 1-in. O.D. (outside diameter) tubes, each with a thickness of No. 18 B.W.G.; the $\frac{3}{4}$-in. tube has the highest rate of transmission. The curves are based upon an inlet circulating water temperature of 70 deg. F. and a steam supply of 8 lb. per sq. ft. per hr. A correction curve "Temperature curve" is given for other water temperatures. The correction curve "Loading curve" is used for partial load conditions; the entire cooling surface, although available, is not effectively used, and to get a mean value the heat transfer must be reduced. If a condenser were designed for 70 deg. F. inlet water, 8 lb. of steam per sq. ft. per hr.; $\frac{3}{4}$-in. tubes, and a velocity of the circulating water through the tubes of 6 ft. per sec., the value of U from Fig. 233 would be 665. If this condenser were supplied with 60 deg. F. water at a velocity of 4.5 ft. per sec. through the tubes and 4 lb. of steam per hr. per sq. ft., the value of U would be $0.92 \times 0.84 \times 575 = 444$.

It is definitely known that the quantity of heat passing through the cooling surface of any heat exchanger is proportional to the temperature difference at any instant. The instantaneous temperature difference is indeterminate, hence it is necessary to establish a *mean temperature difference* for the entire period of thermal contact of steam and circulating water in a surface condenser. The logarithmic or arithmetical mean temperature may be used for this purpose, but the determination of either value is based upon assumptions which do not represent actual conditions. In general,

$$D_m = (t_c - t_1) - (t_s - t_2) \div \log_e \frac{(t_c - t_1)}{(t_s - t_2)}; \tag{138}$$

$$D'_m = \frac{(t_c - t_1) + (t_s - t_2)}{2}; \tag{139}$$

in which D_m = logarithmic mean temperature difference, deg. F.
D'_m = arithmetical mean temperature difference, deg. F.
t_s = temperature in steam space, deg. F.
t_c = temperature of condensate leaving condenser, deg. F.

Other items as in equation 136.

The average temperature in the steam space of a surface condenser

approximates the temperature of the steam entering the condenser, or $t_s = t_c$; hence the A.S.M.E. recommends

$$D_m = (t_2 - t_1) \div \log_e \frac{(t_s - t_1)}{(t_s - t_2)}; \tag{140}$$

$$D'_m = t_s - \frac{(t_2 + t_1)}{2}. \tag{141}$$

Most engineers prefer the logarithmic mean, which should be adhered to for all heat-transfer calculations where the range of temperature is high, because the assumptions more nearly approach actual conditions. However, some authorities have advised the use of the arithmetical mean as sufficiently accurate for purposes of comparison, especially when the rise in temperature of the circulating water is less than 10 deg. F.

268. Comparison of Condensers.—*Jet condensers* have a small initial cost, require less building space, use less cooling water, can use acidulated water, and do not depreciate rapidly, as compared with other types of condensers of equal capacity. Large quantities of dissolved gases enter the condensing chamber of a jet condenser with the injection water, and the liberation of these gases under the reduced pressure tends to decrease the vacuum, that is, increases the exhaust pressure. The additional work required to remove the liberated non-condensable gases, the danger of flooding the prime mover, and the waste of the condensate are other disadvantages of the jet condenser.

The *barometric condenser* is not costly, but the piping to it is expensive and adds to the possibility of air leaks; the loss of vacuum between the steam prime mover and condenser may amount to $\frac{1}{2}$ in. of mercury or more. The condensate is wasted, but danger of flooding the prime mover does not exist as in the low-level jet condenser. The use of acidulated water is possible; operation and maintenance costs are low because the equipment is simple, no hot-well pump being necessary, and some forms of the barometric condenser do not require a vacuum pump. In general, the vacuum obtainable with barometric condensers is slightly higher than with low-level jet condensers. Building space and circulating water requirements are less than for surface condensers of equal capacity.

Surface condensers are installed when high vacua are to be maintained and where the re-use of the condensate as boiler feedwater is essential. The first cost and maintenance expense of surface condensers are high, and considerable building space is required to remove the tubes. The heat transmission of the tubes in a surface condenser is decreased by lubricating-oil deposits from the exhaust steam of a reciprocating engine

or by the deposit of foreign matter from the circulating water. The tubes of surface condensers require frequent cleaning and may show considerable corrosion in short intervals of time. The use of acidulated water or water containing foreign matter in large quantities may preclude the use of surface condensers.

269. Water-cooling Methods.—When an ample supply of water is unobtainable at a low cost or from a natural source, such as a lake or river, some means of cooling the circulating water may be used. The condensing water is often circulated continuously, being cooled after

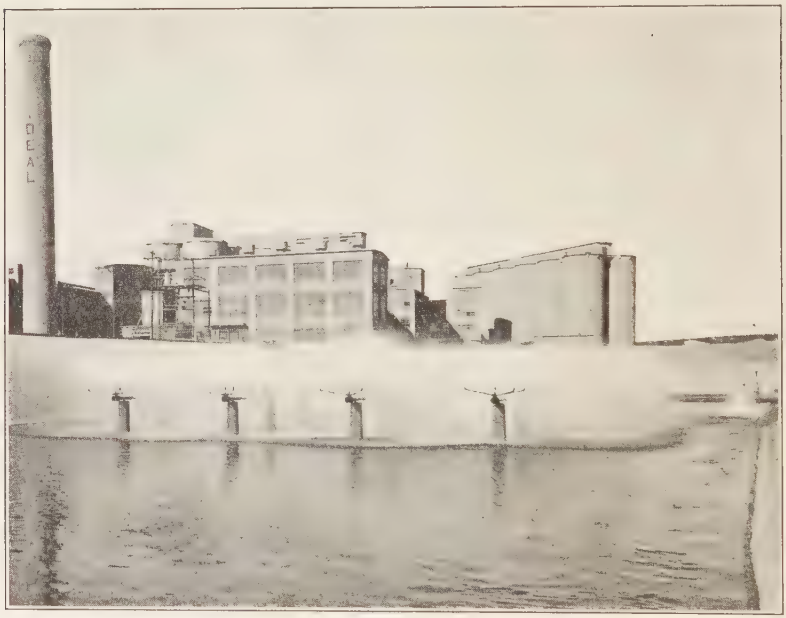


FIG. 234.—Binks spray-cooling system.

each passage through the condenser. The cooling is effected by the exposure of a large surface of the water to the air. The resultant evaporation of some of the water with the absorption of the latent heat of vaporization cools the remainder, so that the cooled water can be used again. This process of recooling the water is usually accomplished by means of cooling ponds or cooling towers.

The oldest means of cooling and storing condenser circulating water is to discharge the hot water into a pond of sufficient area, so that the water is cooled by the air passing over the surface of the pond. This method requires ponds of large size, and in order to reduce the area necessary the heated water may be sprayed into the air over the pond,

Fig. 234, thus increasing the rate of cooling. The cooling action is effected by convection and evaporation. Natural ponds without spray nozzles require 30 to 50 times the area of spray cooling systems.

Because of space requirements and windage loss with the spray pond, the cooling tower has been developed. Cooling towers, Fig. 235, are large, rectangular wood or metal structures generally arranged with some form of baffling devices. The hot water is delivered to the top and falls in thin sheets or sprays into a tank below. Air enters at the

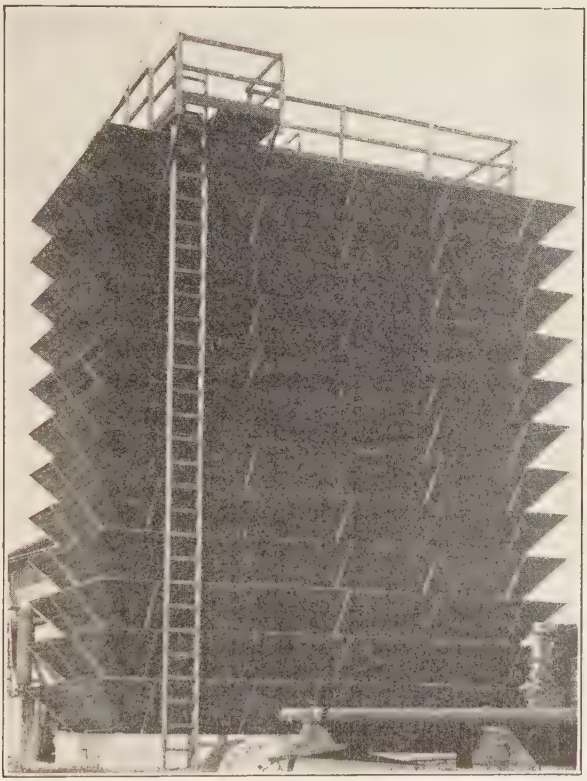


Fig. 235.—Fluor redwood cooling tower.

bottom and flows upward, cooling the water by contact (convection) and by the partial evaporation which results. The circulation of air may be effected by either natural or forced draft. With natural draft, the difference of temperature between the air inside and outside the tower causes the flow, the chimney action may be aided by fitting a stack to the top of the tower. Forced-draft towers have entirely closed sides, except for openings at the base containing one or more fans; the heated and saturated air leaves through the open top of the tower.

PROBLEMS

1. The vacuum gage of a jet condenser showed 25 in. of mercury. Barometer, 29 in. of mercury. Injection water entered at a temperature of 65 deg. F., discharge temperature was 105 deg. F. (*a*) If 10,000 lb. of dry steam entered the condenser per hour, how many pounds of injection water were required per hour? (*b*) How many pounds of water were required per pound of steam?

2. Find the quantity of cooling water theoretically required per pound of dry steam condensed in a jet condenser in which a vacuum of 25 in. of mercury was maintained, when the barometer was 29 in. of mercury. Assume the initial temperature of the water as 65 deg. F.

3. A steam engine delivered 60 brake hp. when operating on 98 per cent quality steam at 120 lb. per sq. in. abs. pressure and exhausting into a jet condenser having a vacuum of 24 in. of mercury. Barometer, 28.85 in. of mercury. How much injection water at a temperature of 70 deg. F. was theoretically required in pounds per hour, if the engine had an economy of 35 lb. of dry steam per brake hp-hr.? Mechanical efficiency, 94 per cent. (NOTE.—Work was done in the engine by part of the heat of the steam.)

4. A steam engine developed 100 i.hp. and required 28 lb. of steam per i.hp-hr; steam was supplied dry and at a pressure of 150 lb. per sq. in. abs. The exhaust was delivered to a jet condenser where the pressure was 2.4 lb. per sq. in. abs. The injection water was supplied at a temperature of 55 deg. F. and discharged at 97 deg. F. Assume no loss due to radiation. (*a*) How many pounds of injection water were required per hour? (*b*) Find the quality of the exhaust steam. (NOTE.—Work was done in the engine by part of the heat in the steam.) (*c*) What would be the quality of the steam if adiabatic expansion were assumed?

5. The steam consumption of a turbine was 18 lb. per kw-hr.; steam pressure, 200 lb. per sq. in. abs.; superheat, 118 deg. F. The vacuum in the jet condenser, serving this turbine, was 27 in. of mercury; injection water entered at a temperature of 57 deg. F. and was discharged at 85 deg. F. Barometer, 29.6 in. of mercury. Determine the weight of water required per pound of steam, if the radiation loss was one per cent of the heat supplied to the turbine.

6. A surface condenser received circulating water at a temperature of 56 deg. F. and discharged it at 74 deg. F. A vacuum of 27.7 in. of mercury was maintained with the barometer at 29.2 in. The temperature of the condensate discharged to the hot well was 83 deg. F. Find the quantity of circulating water theoretically required per pound of steam, if the entering steam contained 10 per cent moisture.

7. A 5000-kw. turbine required 14 lb. of steam per kw-hr. The steam was supplied to the turbine at 400 lb. per sq. in. abs. and a total temperature of 550 deg. F. The steam was exhausted into a surface condenser against an absolute back pressure of 2 in. of mercury. Temperature of circulating water entering condenser, 70 deg. F.; temperature of circulating water leaving condenser, 85 deg. F.; and temperature of condensate, 90 deg. F. How many gallons of circulating water were supplied per minute? Use 8.33 lb. per gallon for water at 70 deg. F.

8. A surface condenser required 60 lb. of circulating water per pound of steam. The circulating water was supplied at a temperature of 70 deg. F. The pressure in the condenser was 3 in. abs., and the condensate temperature was 5 deg. F. below that of the steam in the condenser. What was the probable temperature of the circulating water leaving the condenser, if the quality of the entering steam was 80 per cent? Assume no loss due to radiation.

9. In a surface condenser, the initial temperature of the circulating water was

55 deg. F. and the final temperature 100 deg. F. Dry steam entered an engine at 140 lb. per sq. in. abs., and the engine exhausted into the condenser at 3 lb. per sq. in. abs. The engine developed 200 i.hp. and used 22 lb. of steam per i.hp-hr. The final temperature of the condensed steam was 120 deg. F. (*a*) What was the quality of the steam entering the condenser? (*b*) Assuming no radiation losses, how many pounds of circulating water were required? (*c*) Find the actual thermal efficiency of the engine, based on i.hp.

10. The heat supplied to a surface condenser was 1000 B.t.u. per lb. of steam. Determine the number of square feet of tube surface, from the following data: temperature of exhaust steam, 135 deg. F.; circulating water inlet, 70 deg. F.; circulating water outlet, 100 deg. F.; condensate, 105 deg. F.; U, 650; and weight of steam, 60,000 lb. per hour.

11. A 1000-kw. turbine-generator used 15 lb. of steam per kw-hr. at rated load when operating with steam at 175 lb. per sq. in. abs. pressure and 75 deg. F. superheat with a condenser pressure of 3 in. of mercury abs. Determine the square feet of one-inch tube surface required in a surface condenser for this unit; inlet water at 70 deg. F.; outlet water at 88 deg. F.; and condensate leaving condenser at 5 deg. F. below the saturation temperature. Assume no radiation loss and water velocity through tubes at 6 ft. per sec. Use Fig. 233 to find U.

12. A 25,000-kw. turbine-generator used 11.4 lb. of steam per kw-hr. at rated load. Initial steam pressure, 400 lb. per sq. in. abs.; superheat, 200 deg. F.; and the vacuum in the surface condenser, 29 in. referred to a 30-in. barometer. Temperature of inlet circulating water, 75 deg. F.; outlet water, 90 deg. F.; and condensate, 3 deg. F. less than the steam temperature. Efficiency of the turbine-generator was 90 per cent. The condenser tubes were $\frac{3}{4}$-in. O.D.; thickness, 0.049 in.; water velocity in tubes, 8 ft. per sec. Assuming no radiation loss, find: (*a*) Weight of circulating water required per hour. (*b*) Ratio of circulating water to steam supplied. (*c*) Number of square feet of condenser tube surface. (*d*) Ratio of condensing surface to kilowatt capacity of turbine. (*e*) Number of tubes, two-pass condenser. (*f*) Length of tubes in feet.

CHAPTER XVII

PUMPS

270. Classification.—Pumps, for imparting velocity and pressure to fluids, may be grouped as follows:

1. Piston pumps { Single cylinder / Multicylinder
2. Centrifugal pumps
3. Rotary pumps
4. Fluid-impellent pumps { Continuous flow / Intermittent flow

Piston pumps work by exerting a pressure directly on the fluid being pumped. Centrifugal pumps exert the pressure indirectly by gradually arresting the momentum of the fluid given to it by rapid rotation of an impeller within a suitable casing. The water inlet and discharge of the piston pump are controlled by valves which open and close intermittently, whereas in the centrifugal pump the entry and discharge are continuous with no valves and no pump controlling devices. Rotary pumps, similar to Fig. 251, combine the positive-pressure method of the piston pump with the continuous entry and discharge of the centrifugal pump by employing rotating impellers (or screws).

In fluid-impellent pumps the velocity and pressure of one fluid act directly on another, thereby imparting all or part of its energy to the latter fluid. Injectors, Art. 114, ejectors, aspirators, and some air lifts are of the continuous-flow type. The pulsometer pump, hydraulic ram, pneumatic pump, and the gas-explosion pump are typical intermittent fluid-impellent pumps. These special forms of pumps have their own more or less definite applications, and their use is generally dependent upon special service conditions. Fluid-impellent pumps will not be further discussed as they are not usually in commercial competition with other forms of pumps.

Vacuum pumps for the removal of air and vapors from steam condensers have been described in Chapter XVI.

271. Direct-acting Steam-driven Piston Pump.—A piston or reciprocating pump in its simplest form consists of a cylinder provided with inlet and outlet openings, suitably controlled by valves, and a recipro-

cating piston or plunger working within the cylinder. This type of pump may be used for forcing feedwater into a boiler, removal of condensate and air from a condenser, and in elevator and water-works service.

A steam pump that has no crank or flywheel is known as a *direct-acting pump,* Fig. 236; the water piston and steam piston are fastened to the same piston rod, and the steam pressure is transmitted directly to the water. In this type of pump the steam is not used expansively, and therefore the steam consumption is large. When a single steam cylinder and a single water cylinder are employed the pump is called a simple or *simplex steam-driven pump;* when two steam cylinders and two water cylinders are used, the pump is called a *duplex steam-driven pump.* The size of this type of pump is stated by giving first the diameter of the steam cylinder, next the diameter of the water cylinder, and then the length of the stroke; thus 10 in. by 6 in. by 12 in.

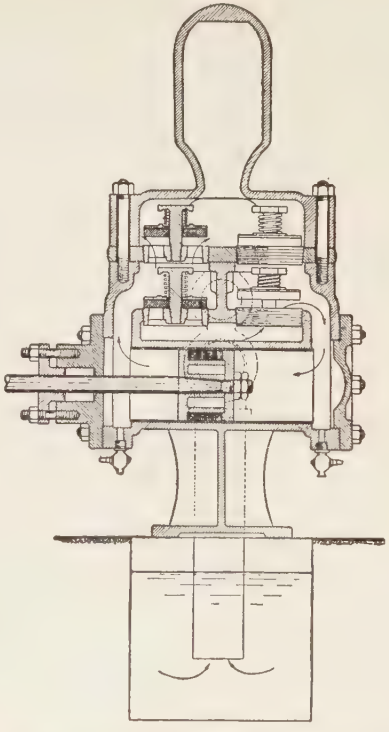


Fig. 236.—Water end of a direct-acting pump.

Direct-acting steam pumps for power-plant service are of the reciprocating double-acting type. Such pumps are designed to raise water by suction from a lower level, and to deliver the water during each stroke of the piston or plunger to a boiler, tank, etc. The movement of the piston toward the left, Fig. 236, forces the water through the discharge valve (upper left), as indicated. Coincidentally, a partial vacuum is created on the right side of the piston, and when the reduction in pressure is large enough, atmospheric pressure forces water up the suction pipe and through the suction valve (lower right), as shown. On the return stroke of the piston the right side discharges water and the left side receives water from the source of supply. The intake water often flows under pressure to the suction pipe, as from an elevated tank or lake, or from street mains.

272. Simple Steam-driven Pump. –The water end of the steam-driven single-cylinder pump, Fig. 237, has the same general construction

as shown by Fig. 236. The steam cylinder and valve chest are practically the same as those described under the steam engine, with the addition of an auxiliary control F of the slide-valve G. The piston C is driven by steam admitted under the slide-valve G. As G is shifted from right to left, it alternately connects opposite ends of the cylinder A with the live-steam pipe and the exhaust. Slide-valve G is actuated by the auxiliary piston F. The hollow ends of F are filled with steam, and this steam, issuing through a hole in each end, fills the spaces between the plunger and the heads of the steam chest. Piston valve F, under ordi-

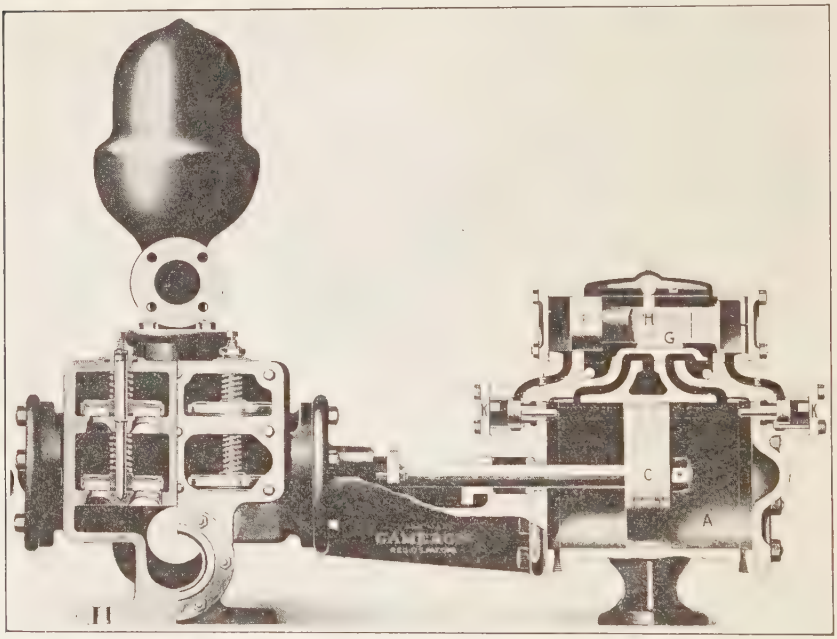


Fig. 237.—Section through a simple direct-acting steam pump.

nary conditions, is balanced and motionless, the pressure being equal at each end. When the main piston C has traveled far enough to the left to strike and open the reverse valve I, the steam flows from the left end of F through port E to the exhaust. Because of this reduction in pressure at the left end of F, the piston valve F is forced to the left by the higher pressure at the right end; the movement of F actuates the slide-valve G, thus reversing the stroke of the piston. In its movement, F acts as a slide-valve to close the port E, and is cushioned by the confined steam between the port and steam-chest cover. As soon as piston C moves out of contact with valve I, the latter is closed by steam pressure

conveyed directly from the steam chest through ports K, as shown by the dotted lines.

The *water valves* of piston pumps are generally of the poppet-disk type rising vertically from flat seats. The pump valves shown in Figs. 236 and 237 consist of a rubber-composition disk supported by a brass cap and are held firmly to the seat by coil springs. For pressures exceeding 200-lb. gage, or for handling hot water, spring-loaded metallic valves are frequently used.

Air chambers attached to the discharge of piston pumps, as in Figs. 236 and 237, provide an elastic element in the pipeline, reduce the shocks and irregularities, and maintain a uniform flow of water in the pipe.

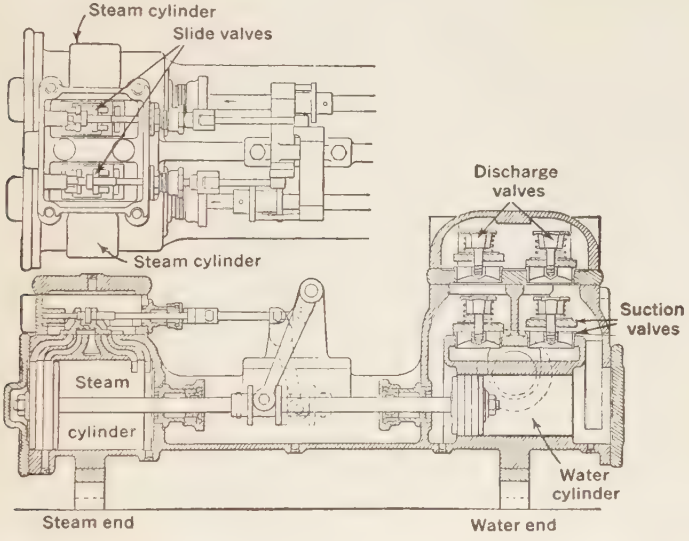


Fig. 238.—Plan and sectional elevation of duplex double-acting steam-driven pump.

The air in the chamber is compressed during each stroke of the water piston; when the piston stops momentarily at the end of the stroke, expansion of the compressed air tends to maintain a uniform rate of discharge. Air chambers are a necessity for simplex pumps, single-acting pumps, and with crank and flywheel pumps. The air chamber may be omitted for small duplex double-acting pumps (Fig. 238), sizes 10 in. by 6 in. by 10 in., or less. Suction air chambers are frequently provided on long- or high-suction lifts for single pumps, on all fire pumps, and any pumps which run at high speeds, particularly those having a short stroke.

273. Duplex Steam-driven Pump.—A duplex pump, Fig. 238, has two steam cylinders and two water cylinders. This pump comprises, in

effect, two single pumps placed side by side, water entering through a common suction pipe and discharging into a common discharge chamber and pipe. The operation of the water end of the duplex pump is similar to that of the water end of the direct-acting pump illustrated by Fig. 236. The steam valve of each cylinder is actuated by the piston rod of the other cylinder, through a system of oscillating rocker arms and bell cranks. The flow of water is practically continuous, as one or the other of the pistons is always in motion.

The slide valves used in the steam end of duplex pumps have no lap. Separate inlet and exhaust ports are provided at each end of the steam cylinder in order to decrease the valve travel and minimize steam leak-

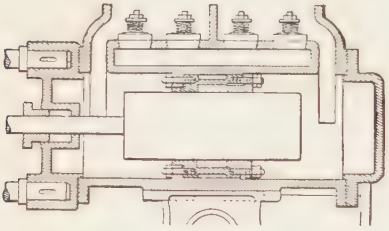


FIG. 239. Pump plunger, inside-packed with fibrous hydraulic packing.

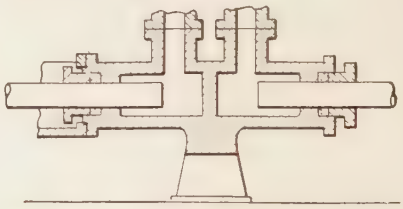


FIG. 240.—Outside end-packed plunger pump.

age, also to cushion the piston near the end of the stroke with the entrapped exhaust steam. There is some lost motion between the slide-valve and its operating mechanism, thus permitting adjustment of the piston stroke and preventing the pump from stopping in a position from which it cannot be started by admitting steam to the steam chest. This arrangement also causes a pause in the flow of water at the end of the stroke, thus giving the water valves an opportunity to seat quietly.

274. Plunger Pumps. The water end of a direct-acting pump is often fitted with a plunger (Fig. 239) instead of a piston (Fig. 238). The plungers pass through fibrous or metal-packed stuffing boxes to prevent leakage, the packing being located either inside or outside of the cylinder. In Fig. 239, the packing is inside the cylinder, thus leakage is not readily detected. The packing is adjusted and replaced by removing the cylinder head. The plungers, Fig. 240, are outside packed at the ends of the cylinder; leakage is visible and may be prevented by adjusting the gland against the packing. These two plungers move together, being connected to an external yoke, the latter is usually actuated by a reciprocating steam engine. Plunger pumps are generally used for discharge pressures above 200 lb. per sq. in.

Piston and plunger pumps which operate at a uniform speed with the aid of the inertia effect of a flywheel are called *flywheel* or *rotative pumps*. When the reciprocating movement of the piston or plunger is derived from the rotation of a mechanically driven crankshaft, the pump is known as a *power pump*. Power pumps may be operated at constant speed by gears, belts, or chains, and the source of power can be steam, gas, or electricity. When one plunger is employed they are called simple or simplex; two plungers, duplex; and three plungers, triplex power pumps.

Figure 241 shows a vertical triplex power pump (single-acting) driven by an electric motor through a set of gears. The rotation of the crank-

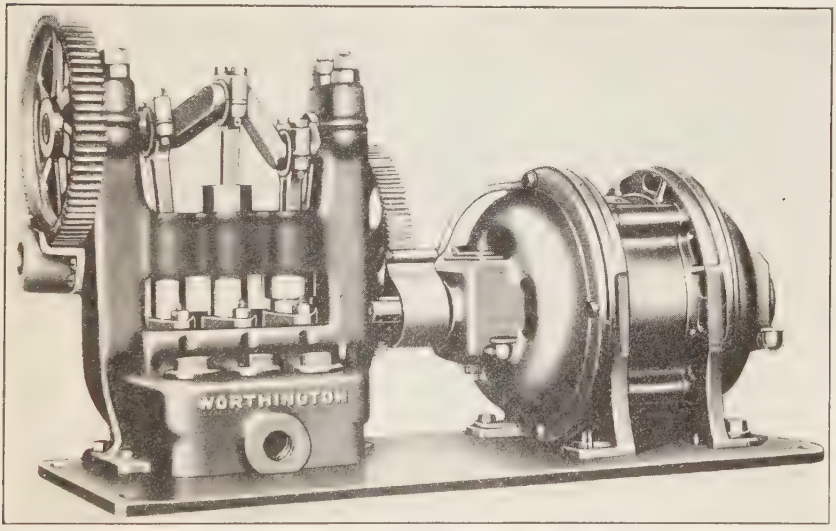


FIG. 241.—Single-acting vertical triplex power (plunger) pump.

shaft is transmitted to each plunger by a connecting rod. The base is made in compartments, and the operation is the same as for the water end of any piston pump. The efficiency of this type of pump depends largely upon the character of the driving unit and the efficiency of the transmitting mechanism. With an electric motor drive the overall efficiency of power pumps is generally between 60 and 80 per cent.

275. Centrifugal Pumps.—The centrifugal pump is a simple mechanism with no valves or springs. In brief, the centrifugal pump consists of an impeller rotating within a casing, as shown in Figs. 242 and 243. Centrifugal pumps are relatively high-speed machines and are generally direct-connected to a steam turbine or an electric motor; belt and rope drives or internal-combustion engines may also be used. Water enters

the impeller, Figs. 242 and 243, at the center (around the shaft), flows radially outward, and leaves the periphery of the impeller at a velocity which is a resultant of the tip velocity of the impeller blade and the radial flow of the water through the pump. This velocity of the water is gradually decreased, and the energy of motion is converted into pressure head in the pump casing in which the impeller rotates. The water is under pressure and will flow out of the pump against the resistance to be overcome. The design of a centrifugal pump is generally directed towards converting the energy of velocity into potential energy without loss, minimizing the friction of disk rotation, and balancing the end thrust of the impeller shaft.

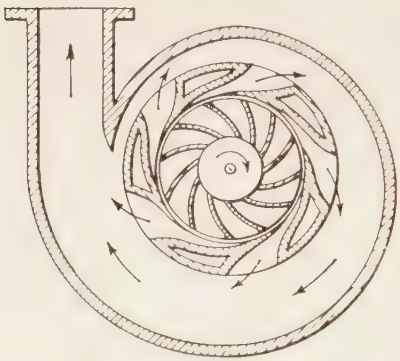


FIG. 242.—Single-stage turbine pump with volute casing.

Centrifugal pumps may be divided into the following groups: (1) *Turbine and volute;* (2) *number of stages;* (3) *single-suction and double-suction;* (4) *open-impeller and enclosed-impeller;* (5) *horizontal and vertical.*

The turbine centrifugal pump has an impeller which is surrounded by a casing containing diffusion vanes (stationary) as shown in Fig. 242. These diffusion vanes provide gradually enlarging areas, thus reducing

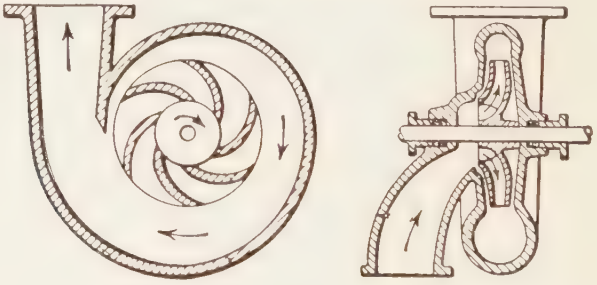


FIG. 243.—Single-stage, single-suction volute centrifugal pump.

the velocity of the water leaving the impeller and changing the velocity head into pressure head. The volute pump, Fig. 243, uses no diffusion vanes, but the casing is of a spiral type forming a gradually increasing water space, thus gradually reducing the velocity of the water as it flows from the impeller to the discharge pipe. Volute pumps ordinarily have but one impeller. The volute pump is considered superior to the

turbine pump from a standpoint of first cost, simplicity, and size, but for discharge pressures greater than 50 lb. per sq. in., the turbine pump usually shows better performance.

The multistage centrifugal pump, Fig. 244, consists of two or more impellers through which the water passes successively. The impellers are mounted on the same shaft and revolve in separate compartments or stages. The discharge from each stage is delivered to the suction of the next higher stage, the final delivery pressure depending upon the number of stages. Multistage centrifugal pumps are generally employed when the discharge pressure exceeds 100 lb. per sq. in. These pumps are usually of the turbine type with single inlet impellers. In

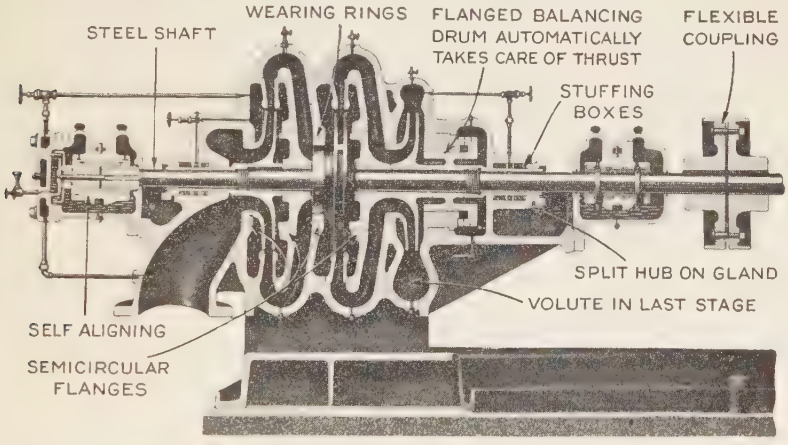


Fig. 244.—Cameron three-stage centrifugal pump.

Fig. 244 the first two stages are of the turbine type and the third stage employs a volute impeller.

In the single-suction centrifugal pump, Figs. 243 and 244, the water enters the impeller from one side only. A double-suction pump, Fig. 245, is one in which the water enters from both sides. For a given impeller diameter, the double-suction pump will discharge more water than a single-suction pump. The side thrust produced by the pressure of the water as it enters the one side of the impeller in the single-suction pump necessitates special provision for counterbalancing this thrust. The counterbalancing may be accomplished by a thrust bearing or by an equalization of pressures (hydraulic balancing) on the two sides of the rotor.

The open-impeller centrifugal pump, similar to Fig. 243, is sometimes called a fan pump. The operation is similar to that of a paddle-wheel revolving in a casing. Poor water-guidance results in excessive eddying

of the water; also a large amount of water escapes between the blades of the impeller and the casing walls. The open-impeller design gives a low efficiency and is used only in the cheaper pumps for handling thick liquids such as mash, pulp, and sewage. The enclosed or shrouded type of impeller, Figs. 242 and 246, consists of blades and side plates forming enclosed water passages. The water enters the passages at the center, and the enclosing walls prevent water from escaping past the impeller blades. Where an enclosed-type impeller is employed a double suction is used thereby minimizing end-thrust on the shaft. Impellers are usually made from bronze, which has proved best suited for with-

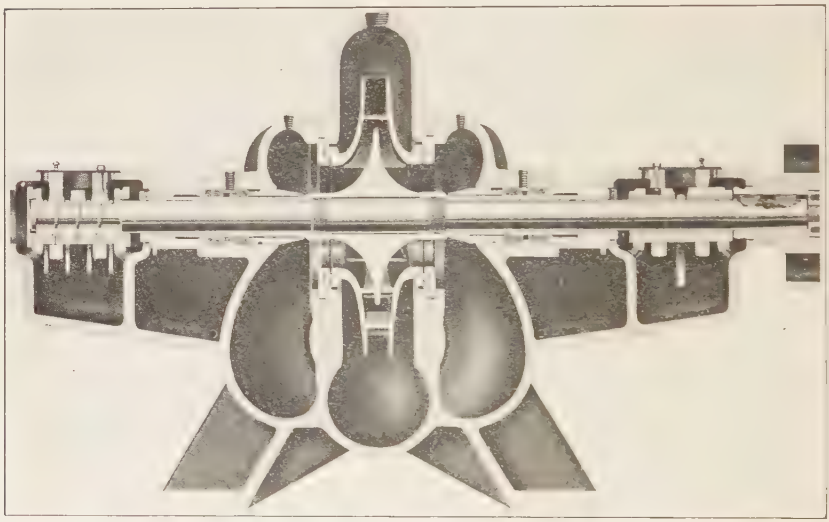


FIG. 245.—Sectional view of De Laval single-stage double-suction centrifugal pump.

standing the action of water and which retains its original shape and smoothness.

A vertical-shaft centrifugal pump may be necessary where conditions are such that a horizontal-shaft pump cannot be placed within suction distance (less than 23 ft.) of the supply water level. This condition is frequently encountered in deep wells, sewage service, sumps, and near rivers where the difference in water level varies considerably. The pump may be completely submerged in the water, but when possible it is located in a dry pit in order to be easily accessible. The bearings of vertical pumps require considerable attention, and hence the horizontal-shaft type is used wherever feasible.

276. Comparison of Pumps. -*Piston pumps* have a wide field of application in small power plants and for certain industrial purposes,

especially where the quantity of fluid to be handled is small and the total pressure pumped against is high. The advantages of a piston pump are that the action is positive, the pump does not "lose its water," and it will generally operate when in a very indifferent state of repair. The pump input or power required can readily be determined; the speed alone governs the quantity of water pumped. Where either the head or the discharge (quantity) is required to vary within wide limits and where they do not maintain definite relations to each other, the displacement pump will be found to be flexible and economical. Direct-

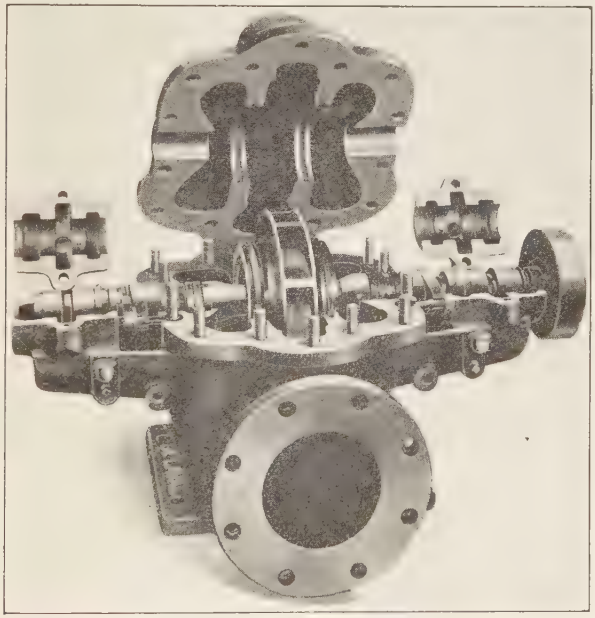


FIG. 246.—Single-stage double-suction centrifugal pump with casing cover and bearing caps lifted.

acting steam pumps are very wasteful of fuel and are low in thermal efficiency, largely because of the non-expansive use of steam and slow-speed operation. The simple or simplex pump generally uses less steam for the same amount of work than does the duplex pump on account of smaller clearance spaces in the steam cylinder of the former type. The piston of the simple pump travels the maximum set distance during each stroke, whereas the pistons of the duplex pump may "short-stroke" owing to the retarding effect of friction between the pistons and cylinder walls and friction in the piston-rod stuffing boxes. Simple pumps are well adapted for vacuum pumps in connection with condensers as the steam cylinder clearances are comparatively small and

there is no short-stroking. The practically continuous action of duplex pumps makes them better adapted for high-pressure service than simple pumps; the former are also preferable where either a very high or a very low velocity of flow is required.

Flywheel and power pumps as compared with direct-acting steam pumps have a higher overall efficiency, may be run at higher speeds, there is no danger of short-stroking, and less clearance is necessary. The first cost of direct-acting pumps is less, also the attendance and maintenance required are less than for flywheel and power pumps. However, power pumps may be located many miles from the source of power, and automatic control may be effected readily when they are electrically driven.

The lack of balance of piston and plunger pumps necessitates costly foundations. The repeated reversals of flow of the fluid cause stresses which may shatter the pump or its connections. These pumps are bulky and costly for a given output, and often the extra interest on the first cost and maintenance offsets any benefits from a probable higher efficiency. Without a flywheel, piston pumps are difficult to drive by belt, gas engine, or electric motor. Piston and plunger pumps cannot satisfactorily handle dirty, gritty, and highly viscous fluids.

The *centrifugal pump* with steam-turbine or electric-motor drive combines maximum efficiency in power transmission with simplicity, compactness, balanced rotary motion, absence of valves and pistons, uniform discharge, reliability, and small weight. For pumping small volumes of water against high pressures, the centrifugal pump can be connected directly to, and run at the same speed as, the turbine. Where the proper speeds of the pump and turbine are too far apart to permit satisfactory efficiency at a compromise speed, or where the quantity of water to be handled is very large, speed-reducing gears are interposed between the turbine and pump. Boiler-feed pumps, condensate pumps, hydraulic-pressure pumps, elevator pumps, fire pumps, mine pumps, and other similar-purpose pumps are usually directly connected to the turbine. Condenser-circulating pumps, large general water-service pumps, and water-works pumps are generally driven by gears. The centrifugal pump discharges a continuous stream of fluid. Originally developed as a low-lift pump, the centrifugal pump is now used for high-pressure service, and pressures exceeding 850 lb. per sq. in. can be obtained in a 4-stage unit. This type of pump is less costly to install, operate, and maintain than the piston type, and will pump water containing mud and small stones. Some of the disadvantages of centrifugal pumps are: the efficiency is not usually as high as that of the best grade of piston pump, and the pump is not satisfactory where a high-suction lift is necessary or for pumping small quantities of water against

high discharge pressures. A centrifugal pump when designed for a certain capacity and total pressure cannot be used without considerable loss of efficiency at any other capacity or pressure. The reciprocating pump is more flexible in this respect, as it can be operated under widely different conditions without any large sacrifice of efficiency.

277. Performance of Pumps.—Suitability of a pump for a given service is based upon the relations of head, capacity, speed, power, and efficiency. The relation between the fundamental variables, head and capacity, may be represented by means of a curve; the curves for efficiency and brake horsepower may also be plotted. These curves are based on actual test results and vary with the design of the pump; when taken together they form the "characteristics" of the pump. Other curves may be obtained, however, by keeping any one of the fundamental quantities constant and varying the others.

The *total head* against which a pump is delivering water is equal to the sum of the change of velocity head in the pump and the difference between the head with which the water leaves the pump and that with which the water enters the pump.

$$H_t = H_d - H_s + \frac{V_d^2 - V_s^2}{2g}, \tag{142}$$

where H_t = total head, ft. of water.
H_d = discharge head, ft. of water.
H_s = suction head, ft. of water.
V_d = velocity of water in discharge pipe, ft. per sec.
V_s = velocity of water in suction pipe, ft. per sec.
g = acceleration due to gravity, 32.17 ft. per sec.^{2}

In the above equation the small losses in the pump valves are neglected.

Suction lift is the vertical distance from the level of the water to be pumped to the center line of the pump. When the pump lifts the water from a level below the center line, the pressure in the suction line is less than atmospheric and H_s is considered negative. If this water level is above the center of the pump, the pump then receives water under pressure and H_s is considered positive. Discharge head is the vertical distance between the center line of the pump and the level (or its equivalent) to which the water is elevated. The friction head may be determined by the use of charts or tables, to be found in any standard handbook or textbook on hydraulics, which give the frictional losses in pipes, fittings, and valves. These frictional losses are necessarily approximate and depend on such factors as smoothness of pipe, fluid handled, length of time in service, type of valves, radius of bends, etc. The velocity head, sometimes known as the head due to velocity, is the

equivalent head through which the water would have to fall in order to acquire the velocity at a given point, that is, $H_v = \frac{V^2}{2g}$. The change of velocity head $\frac{V_d^2 - V_s^2}{2g}$ should be considered, however, when the discharge pipe is of different size (generally smaller) than the suction pipe, when the total head is low, and when the suction lift is high, or in accurate testing.

Example.—The diameter of the discharge pipe of a boiler-feed pump is 4 in. and the suction pipe is 6 in. The water delivered is 250 g.p.m., and the temperature is 60 deg. F. The discharge pressure near the pump is 190 lb. per sq. in. gage, and the suction is 12 in. of mercury vacuum. Find the total head pumped against if the discharge gage is located 3 ft. above the center of the pump and the vacuum-gage *connection* to the suction pipe is 2 ft. below the center of the pump. Assume that the head necessary to overcome the frictional resistance through the pump is 2.20 ft.

Solution.—Weight of water at 60 deg. F. = 62.37 lb. per cu. ft., from Table XXI; see Appendix.

$$H_d = \frac{190 \times 144}{62.37} + 3 = 441.7 \text{ ft., discharge head.}$$

$$H_s = \frac{-12 \times 13.59}{12} - 2 = -15.59 \text{ ft., suction head.}$$

$$\text{Quantity flowing} = \frac{250 \times 231}{1728 \times 60} = 0.557 \text{ cu. ft. per sec.}$$

From the equation, $Q = AV$, that is, quantity = area $\times$ velocity,

$$V_d = \frac{\text{cubic feet per second}}{\text{area of 4-in. pipe}} = \frac{0.557}{0.0873} = 6.38 \text{ ft per sec.}$$

$$V_s = \frac{\text{cubic feet per second}}{\text{area of 6-in. pipe}} = \frac{0.557}{0.1963} = 2.84 \text{ ft. per sec.}$$

Note: Areas from Table XXII.

Substituting in (142)

$$H_t = 441.7 - (-15.59) + 2.2 + \frac{(6.38)^2 - (2.84)^2}{2 \times 32.17};$$

$$H_t = 460 \text{ ft. of water.}$$

The *capacity* of a pump is generally stated as the quantity of water the pump will discharge, expressed in gallons per minute. For piston pumps, the capacity is usually given as the piston displacement in cubic feet per minute. For a centrifugal pump the quantity of water discharged varies directly as the speed. *Pump slip* is the difference

between the displacement or theoretical discharge of a pump and the actual discharge, and is commonly expressed as a percentage of the displacement. Average values of pump slip range from 3 to 25 per cent. The *volumetric efficiency* of a piston pump is the ratio of the volume of water actually delivered and the displacement of the pump.

The hydraulic or *water horsepower* of a pump is the power actually delivered to the water, as computed from the weight of water actually discharged and the total head.

$$\text{w.hp.} = \frac{w \times H_t}{33{,}000}, \tag{143}$$

in which w = water delivered by pump, lb. per min.
H_t = total head, ft. of water.

The *mechanical efficiency* of a pump is the ratio of the energy converted into useful work to the energy supplied to the pump; that is, the ratio of the water horsepower output to the shaft horsepower input to the pump, or

$$e_m = \frac{\text{w.hp.}}{\text{s.hp.}}. \tag{144}$$

The shaft (or brake) horsepower is considered as the power supplied to the pump shaft by motor, turbine, or pulley.

The economy of a steam-driven pump may be expressed either as the thermal efficiency or as the duty. The *thermal efficiency* is the ratio of the heat equivalent of the water horsepower to the heat chargeable to the steam end. The *duty* of a steam-driven pump is the number of foot-pounds of work done by the pump per million B.t.u. chargeable to the steam end. The term duty may also be applied to a motor-driven pump if the work done per million B.t.u. supplied to the motor is determined. Duties in excess of 150,000,000 ft-lb. have been obtained with turbine-driven centrifugal pumps.

$$e_t = \frac{2545 \times \text{w.hp.}}{W_s(h_1 - h_{f.2})}. \tag{145}$$

$$Duty = \frac{60 \times w \times H_t}{W_s(h_1 - h_{f.2})} \times 1{,}000{,}000. \tag{146}$$

w and H_t as in equation 143.
W_s = weight of steam supplied, lb. per hr.
h_1 = total heat of steam supplied, B.t.u. per lb. above 32 deg. F.
$h_{f.2}$ = heat of the liquid at exhaust pressure, B.t.u. per lb. above 32 deg. F.

Duty may also be defined as the foot-pounds of work done by the pump per thousand pounds of dry steam supplied to the engine. The specifications usually state at what pressure the steam must be supplied.

Example.—A 10-in. by 6-in. by 10-in. direct-acting duplex pump with $1\frac{1}{2}$-in. piston rods makes 35 double strokes per cylinder per minute and delivers 135 g.p.m. of 100 deg. F. water against a total head of 460 ft. The steam is supplied to the steam end at a pressure of 150 lb. per sq. in. gage and exhausts to the atmosphere. Amount of steam supplied is 3800 lb. per hr. at 98 per cent quality. Barometer, 29.6 in. mercury. Assume that piston makes full stroke. Find (*a*) the piston displacement of the pump in cubic feet per minute, (*b*) pump slip, (*c*) water horsepower delivered, and (*d*) the duty of the pump.

Solution.—

$$(a)\ \textit{Piston displacement} = \frac{0.7854\,[6^2 + (6^2 - 1.5^2)]}{144} \times \frac{10}{12} \times 35 \times 2,$$

$$= 22.15 \text{ cu. ft. per min.}$$

$$(b)\ \textit{Volumetric efficiency} = \frac{135 \times 231}{1728} \times \frac{1}{22.15} \times 100 = 81.5 \text{ per cent.}$$

$$\textit{Pump slip} = 100 - 81.5 = 18.5 \text{ per cent.}$$

$$(c)\ \textit{Water horsepower} = \frac{460 \times 135 \times 231 \times 62}{1728 \times 33{,}000} = 15.6.$$

$$(d)\ \textit{Duty} = \frac{460 \times 135 \times 231 \times 62 \times 60}{1728 \times 3800\,[338.2 + (0.98 \times 856.7) - 179.3]} \times 10^6,$$

$$= 8{,}150{,}000 \text{ ft-lb. per million B.t.u.}$$

The characteristic performance curves, see Figs. 247 and 248, for pumps are usually based upon a constant speed of the pump, the curves showing the relation between (*a*) head and capacity, (*b*) shaft horsepower and capacity, and (*c*) mechanical efficiency and capacity. The values used in plotting the above curves are those computed for a given definite speed of the pump, usually that at which the pump operates at the best efficiency or that at which it was designed to operate. For centrifugal pumps, the relation between the various quantities is largely controlled by the angles and the curvatures of the impeller blades, and the shape of the volute, or the arrangement and the design of the diffusion vanes. If the vanes are radial or inclined forward in the direction of rotation, the head will increase with increased capacity; if the vanes are curved backwards sufficiently, the head will remain constant or decrease as the capacity increases, as shown by Figs. 247 and 248. Centrifugal pump and fan characteristic curves are similar. In general, for a given centrifugal pump, the *quantity of water delivered varies directly with the speed*, the *head with the square of the speed*,

and the *power with the cube of the speed.* Thus doubling speed of a pump impeller doubles the quantity of water pumped, produces a head four times as great, and requires eight times as much power to drive the pump.

Since manufacturers furnish curves for their specific output, and as pump-performance curves vary within wide limits, general curves are without purpose except for general approximations. In Fig. 247, the highest point on the total head curve is 820 ft., hence the pump could not be used for a greater head, as at the existing constant speed no water would be delivered. The pump under consideration delivered 1000 gallons of water per minute against a total head of 680 ft. when

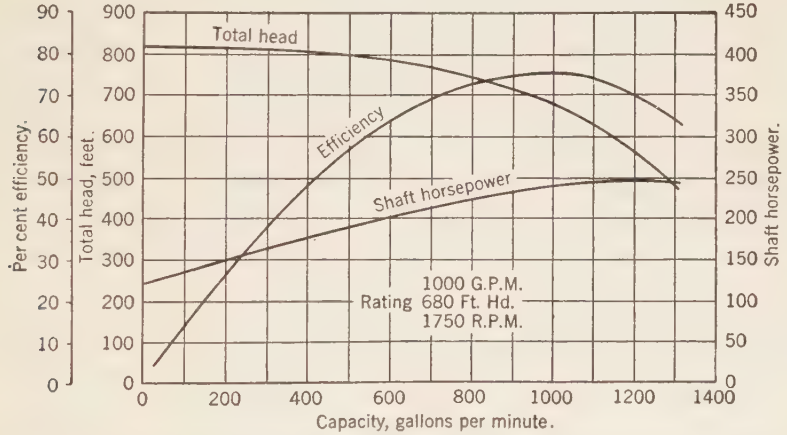


Fig. 247.—Characteristic curves of a Worthington 6-in. by 5-in. (17-in. dia. impellers) three-stage centrifugal pump used for boiler feedwater.

operating at a speed of 1750 r.p.m. Under the conditions mentioned, 238 s.hp. were necessary to drive the pump, and the mechanical efficiency was 75 per cent. Figure 248 shows the performance curves for a high-capacity low-head single-stage centrifugal pump; this pump delivered 5000 g.p.m. against a head of 28 ft. with 39 s.hp. and a mechanical efficiency of 89 per cent when operating at 690 r.p.m. The efficiency curve in Fig. 248 has a gradual curvature between 3000 and 6500 g.p.m.; this is a desirable characteristic, as it shows that the pump can be operated over a wide range of capacity with nearly uniform efficiency. The upper part of the power curve rises gradually, thus additional capacity can be obtained for a relatively small increase in power without sacrificing efficiency but with a reduced head.

278. Measurement of Water.—Methods of water measurement are divided into primary and secondary, according to whether the method

is an absolute measurement of quantity or merely the measurement of an effect of quantity or rate of flow. Of these methods the only primary methods are those of measurement by volume or weight. Secondary methods are more generally used in practice, and include measurement by Venturi tube, weir, orifice,[1] nozzle,[1] or pitot tube.[1]

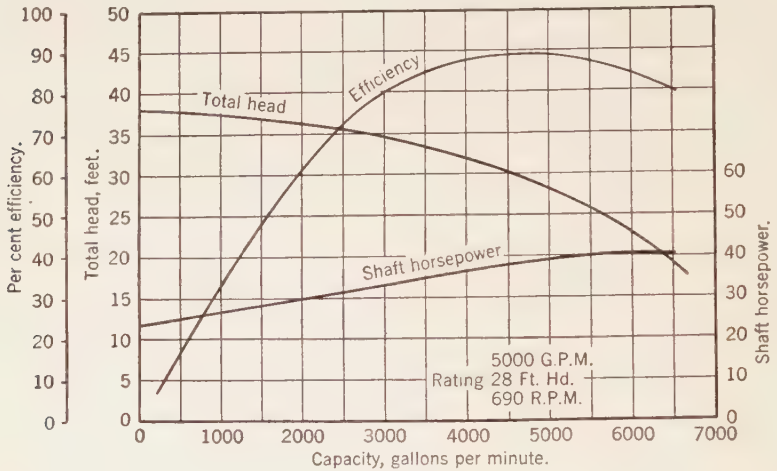


Fig. 248.—Characteristic curves of an Allis-Chalmers 14-in. by 14-in. single-stage double-suction centrifugal pump used for condenser circulating water and general low-pressure service.

For the measurement of flow in pipes under pressure the *Venturi tube*, Fig. 249, is a reliable form of meter and is extensively used where accurate and consistent results are desired. The tube takes the shape of two truncated cones, joined at their smallest diameters by a short throat section, and inserted between flanges in a pipeline. Suitable pressure connec-

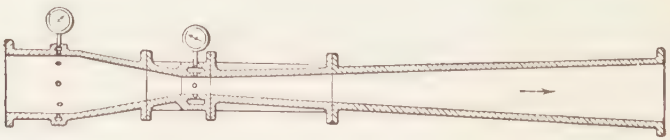


Fig. 249.—Section through a Venturi tube.

tions are provided for observing the pressure at the entrance and at the throat of the Venturi tube. The entrance cone has a total angle of about

[1] For additional information concerning these methods refer to: "Fluid Meters, Their Theory and Application," Part I, Third Edition, 1931; and "Description of Various Types of Commercial Fluid Meters," Part II, 1931; A.S.M.E. Research Publications.

21 deg. and leads to the short cylindrical throat section which is accurately machined. The diameter of the throat is usually between one-fourth and one-half of the pipeline diameter. The exit end of the throat leads to the exit cone or diffuser which has a total angle of 6 deg. The maximum discharge through a Venturi tube is obtained when the throat pressure is 53 per cent of the entrance pressure.

The equation [2] for determining the quantity of water flowing through a Venturi tube is

$$Q = \frac{CA_1\sqrt{2gH_w}}{\left[\left(\frac{D_1}{D_2}\right)^4 - 1\right]^{1/2}}; \qquad (147)$$

in which Q = rate of discharge, cu. ft. per sec.
C = coefficient, usually 0.96 to 0.99.
A_1 = area of pipe at upstream pressure connection, sq. ft.
g = acceleration due to gravity, 32.17 ft. per sec.^{2}
H_w = difference in head, corresponding to the difference between the static pressures at the two "piezometer" connections, ft. of water.
D_1 = diameter of pipe, upstream.
D_2 = diameter of pipe, throat.

The coefficient C varies with the rate of flow, diameter of the pipe, smoothness of inside surface, viscosity of the fluid, and other factors. In practice the numerical value of C is determined from experimental tests.

Example.—The inlet diameter of a Venturi tube is 6 in. and the throat diameter is 2 in. Water flowing through the tube has a temperature of 150 deg. F., an inlet static pressure of 160 lb. per sq. in., and a throat static pressure of 96 lb. per sq. in. How many gallons per minute is the pump delivering? Assume coefficient as 0.99.

Solution.—

$$H_w = \frac{(160 - 96)\,144}{61.2} = 150.6 \text{ ft. of water.}$$

From equation 147, $Q = \dfrac{0.99 \times 0.1963\sqrt{2 \times 32.17 \times 150.6}}{[(\frac{6}{2})^4 - 1]^{1/2}}$;

$$Q = \frac{19.13}{8.95} = 2.14 \text{ cu. ft. per sec.}$$

or

$$= \frac{2.14 \times 60 \times 1728}{231} = 961 \text{ g.p.m.}$$

[2] For derivation, see "Treatise on Hydraulics," Mansfield Merriman, p. 89, Tenth Edition, 1916; John Wiley and Sons, Inc.

The *weir* is an accurate means of determining the rate of flow of large quantities of water in open conduits. There are four principal types: the square notch; the rectangular weir; the V-notch, Fig. 250, which increases the head readings at low rates of flow; and the parabolic notches, which are so shaped as to give head readings directly proportional to the flow. The rectangular weir is generally used for high heads, such as may be found in the measurement of large streams. When the heads are relatively low (less than 12 in.), as in most power-plant service, the V-notch is more reliable. In the weir the variable area of the stream bears a definite relation to the velocity and is indicated by the height of the level of the measured liquid over a horizontal plane taken as a base.

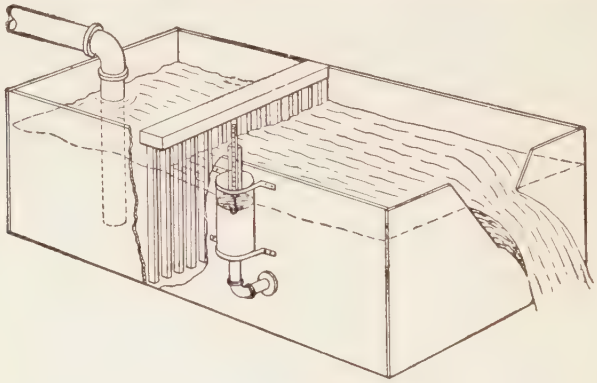


FIG. 250.—V-notch weir with combing rack and hook gage.

This variable height is known as the head, H_w, of the weir and is the difference of elevation in feet between the surface of the liquid in the channel at a point placed just ahead of the beginning of the surface curve and the bottom or sill of the notch. The head of water passing over the V-notch, Fig. 250, is measured by a hook gage which is located at a point where flow occurs without turbulence and ahead of the beginning of the surface curve. The turbulence of the water flowing through a weir tank may be minimized by properly locating suitable baffles; the combing rack, Fig. 250, serves as an efficient means of straightening the flow. For a rectangular-notch weir [3] with two contractions and heads ranging from 3 in. to 24 in., experiments have shown that

$$Q = \tfrac{1}{3}^{0}(B - 0.2H_w)H_w^{3/2}, \qquad (148)$$

where Q = quantity of flow, cu. ft. per sec.
B = base of rectangular notch, ft.
H_w = head of water passing over the weir, ft.

[3] "Treatise on Hydraulics," Mansfield Merriman, p. 154 (rectangular notch), and p. 169 (V-notch), Tenth Edition, 1916; John Wiley and Sons, Inc.

For a triangular weir or V-notch, apex down, with angle at the apex of 90 deg., the rate of flow is

$$Q = 2.53 H_w^{5/2}. \tag{149}$$

Valuable engineering data [4] in the form of accurately determined constants for V-notch weirs are available. Weir equipment of this type is in use at the Richmond Station of the Philadelphia Electric Company as a test meter for the calibration of various orifices and other types of water meters employed in general power-plant service.

PROBLEMS

1. The pressure gage connected to the discharge pipe of a centrifugal pump read 125 lb. per sq. in. and the suction gage recorded a vacuum of 10 in. of mercury. The center of the discharge gage was 18 in. above the center of the pump while the vacuum-gage connection to the suction pipe was 2 ft. below the center of the pump. The suction and discharge pipes were of the same size; temperature of water, 100 deg. F. Neglecting friction between the gages, what was the total head in feet of water? See Table XXI for density of water.

2. A mercury manometer used for measuring the discharge pressure of a rotary pump read 18 in. of mercury; the top of the lower column of mercury in the manometer was 6 in. below the center of the pump when the above reading was observed. The suction gage recorded a positive pressure of one pound per square inch; the suction gage was one foot above the center of the pump. The internal diameter of the suction pipe was 3 in., and that of the discharge pipe $2\frac{1}{2}$ in. If the rate of discharge was 25 g.p.m. and the water temperature 150 deg. F., find the total head developed. Add 3 per cent to the calculated total head for frictional losses between gages.

3. The diameter of the water piston of a direct-acting simplex pump was 6 in. and that of the piston rod was $1\frac{3}{4}$ in. The stroke was 12 in., the volumetric efficiency 95 per cent, and the piston made 40 single strokes per minute. How much water was discharged in gallons per minute?

4. A 12-in. by $7\frac{1}{2}$ in. by 12-in. duplex piston pump with 2-in. piston rods made 30 double strokes per cylinder per minute. The pump delivered 225 g.p.m.; find the volumetric efficiency and the pump slip. The average length of stroke during the test was $11\frac{1}{2}$ inches.

5. A centrifugal pump delivered 800 g.p.m. at 1750 r.p.m. against a head of 750 ft. of water. (*a*) What would be the probable capacity and head at a speed of 1200 r.p.m.? (*b*) What would be the probable speed and discharge for a head of 500 ft. of water?

6. A power pump delivered 4500 g.p.m. of water at a temperature of 60 deg. F. against a total pressure of 50 lb. per sq. in. Find the water horsepower.

7. The internal diameter of the discharge pipe of a turbine-driven centrifugal pump was 10 in., and that of the suction pipe was 12 in. A pressure gage connected to the discharge pipe read 300 lb. per sq. in.; the center of the pressure gage was 40 in. above the center of the pump. A vacuum gage connected to the suction pipe

[4] "Accuracy of the V-notch Weir Method of Measurement," D. Robert Yarnall; Trans. A.S.M.E., Vol. 48.

read 6 in. of mercury; the vacuum-gage connection to the suction pipe was 40 in. below the center of the pump. If under the above conditions this pump delivered 3.27 cu. ft. of water per second with a shaft horsepower of 330, calculate the mechanical efficiency of the pump. The temperature of the water was 70 deg. F.

8. A duplex direct-acting boiler feed pump used 120 lb. of dry steam per i.hp-hr.; i.hp. of steam end, 100; water delivered, 805 g.p.m.; temperature of water, 195 deg. F.; total head pumped against, 160 lb. per sq. in.; steam supply pressure, 135 lb. per sq. in. gage.; moisture in steam, 2 per cent; exhaust pressure, atmospheric; and barometer, 28.88 in. of mercury. Calculate (*a*) the water horsepower, (*b*) the thermal efficiency, (*c*) the duty per million B.t.u. chargeable to the engine, and (*d*) the duty per thousand pounds of dry steam supplied to the engine.

9. A motor-driven centrifugal pump delivered 8000 gallons of water per minute against a discharge pressure of 40 lb. per sq. in. with a suction pressure of 12 in. of mercury, both pressures referred to the center of the pump. The mechanical efficiency of the pump was 78 per cent. What was the horsepower required to drive the pump? Water temperature, 60 deg. F.

10. A centrifugal pump direct-connected to a 25-hp. electric motor was used to handle crude oil at 70 deg. F. and had a capacity of 1500 g.p.m. (*a*) Find the power required to drive the pump if the specific gravity of the oil was 0.87 and the total head was 22 lb. per sq. in. (*b*) What was the mechanical efficiency of the pump?

11. (*a*) Determine the total head in feet of water on a pump that handled brine having a temperature of 50 deg. F. and a specific gravity of 1.25. A discharge pressure of 125 lb. per sq. in. was indicated by a Bourdon pressure gage whose center was 18 in. above the center line of the pump. The suction gage, which was also a Bourdon gage, indicated a vacuum of 12 in. of mercury and was attached to the suction pipe at a point 8 in. below the center line of the pump. (*b*) If 300 gallons of brine were pumped per minute and the mechanical efficiency of the motor-driven pump was 65 per cent, find the brake horsepower output of the motor.

12. A pumping engine had a duty of 20,000,000 ft-lb. per million B.t.u. when discharging 720,000 gallons of water in 24 hours. The temperature of the water was 65 deg. F. The steam pressure was 165 lb. per sq. in. abs. and quality was 98 per cent; exhaust pressure, 8 in. of mercury abs.; discharge pressure of pump, 190 lb. per sq. in.; vacuum gage connected to suction pipe read 10 in. of mercury; and the distance between center of pressure gage and point of attachment of vacuum gage was 4 ft. If the mechanical efficiency of the pump was 80 per cent, determine the dry steam consumption of the engine per brake horsepower per hour.

13. A 10-in. by 6-in. by 12-in. duplex direct-acting boiler feed pump with $1\frac{3}{4}$-in. piston rods made 25 double strokes per minute per cylinder and received water in the suction pipe at a pressure of 3 lb. per sq. in. referred to the center of the pump, and at a temperature of 175 deg. F. The water was pumped through an economizer and entered the boiler at a temperature of 260 deg. F. and a pressure of 135.5 lb. per sq. in. gage (barometer, 29.58 in. of mercury). The pressure on the discharge side of the pump, referred to center, was 150 lb. per sq. in. (*a*) How many pounds of water did the pump deliver per hour, if the volumetric efficiency was 82 per cent? Use weight of water at the suction temperature. (*b*) How many boiler horsepower were being developed in the boiler? (*c*) Find the water horsepower delivered. (*d*) If the combined hydraulic and mechanical efficiency of the engine-pump unit was 75 per cent and the dry steam supplied to the engine from the above boiler was 220 lb. per i.hp-hr., find the duty per million B.t.u. chargeable to the engine, if the engine was operated non-condensing with atmospheric exhaust.

14. The diameter of the water piston of a direct-acting simplex pump was 10 in. and that of the piston rod was 2 in. The stroke was 12 in., and the piston made 64 single strokes per minute. The water was discharged at 160 lb. per sq. in. through a 3-in. by $1\frac{1}{4}$-in. Venturi tube where the pressure at the throat was 88 lb. per sq. in. Assume Venturi tube coefficient as 0.98. Calculate the slip of the pump as a percentage of the piston displacement. The temperature of the water was 120 deg. F.

15. The diameter of the water pistons of a direct-acting duplex pump was 8 in. and that of the piston rods was $1\frac{1}{2}$ in. The stroke was 12 in., and there were 22 double strokes per minute per cylinder. The water delivered flowed over a 90 deg. V-notch weir, the head on the weir being $5\frac{3}{4}$ in. What was the volumetric efficiency of the pump?

CHAPTER XVIII

COMPRESSED AIR

279. Compressor Classification: Uses of Compressed Air.—When large volumes of air are delivered at low pressures, less than 10 in. of water, *fans* are generally employed (see Chapter X); the air may be used for ventilation, combustion of fuel in boiler furnaces, and similar purposes. Air pressures up to 12 lb. per sq. in. gage are produced by *rotary blowers*, as used for supplying air for cupola furnaces, or air to oil and gas burners. The *centrifugal compressor* or turbo-compressor, which is somewhat similar to a centrifugal pump, may be used for air pressures up to 20 lb. per sq. in. gage with a single-stage unit and for higher pressures, up to 125 lb. per sq. in. gage, with the multistage types. Centrifugal compressors may be used in furnishing an air blast for smelting furnaces, Bessemer converters, gas producers, and in "boosting" gas for high-pressure distribution. The *reciprocating piston compressor* is usually employed for producing higher pressures. If the final air pressure is to exceed 100 lb. per sq. in. gage, two or more stages of piston compression are usually employed.

Where power is required intermittently at a distance from its source, compressed air is particularly valuable because the air pressure can be maintained nearly constant during intervals of work. Compressed air may replace steam when power is to be transmitted over a considerable distance, because the steam condensation loss would be excessive, as exemplified in the use of compressed air for mining, air-brake service, sand blasting, and tunneling operations.

280. Rotary Compressors.—For pressures up to 12 lb. per sq. in. gage, and capacities from 5 to 50,000 cu. ft. per min., *rotary blowers* such as that shown by Fig. 251 are used. Two impellers mounted on parallel shafts rotate in opposite directions, drawing air into the pockets between the impellers and the casing, carrying it to the outlet, and discharging it positively. The air is moved at low velocities through free inlet and discharge openings, for the pressure is dependent on the positive mechanical action of the impellers rather than handling the air at high velocities. Loss due to leakage of air past the impellers is very small, resulting in high volumetric efficiencies, because the internal parts fit closely. There is no actual contact between the

impellers and the casing, or the impellers themselves. Blowers of this type are used as aids to combustion in oil and gas burners, foundry cupolas, etc.; for aeration and agitation processes, pneumatic conveying, mine and tunnel ventilation, and many other applications requiring the delivery of a definite volume of air at the pressure required. Rotary positive blowers may also be used to produce a partial vacuum up to 15 in. of mercury.

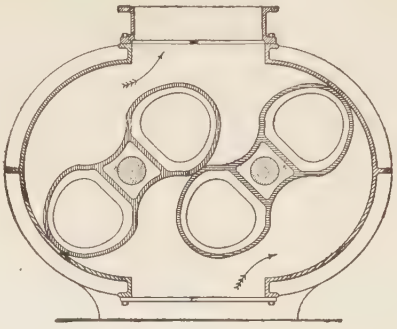


Fig. 251.—Interior view of R.C.W. rotary positive blower.

Centrifugal compressors, also called turbo-compressors, are used for air pressures of 10 to 20 lb. per sq. in. gage with a single-stage unit. For pressures up to 125 lb. per sq. in. gage the compression is accomplished in several stages; the pressure obtained depends primarily upon the speed and the number of stages or impellers used. Figure 252 shows a

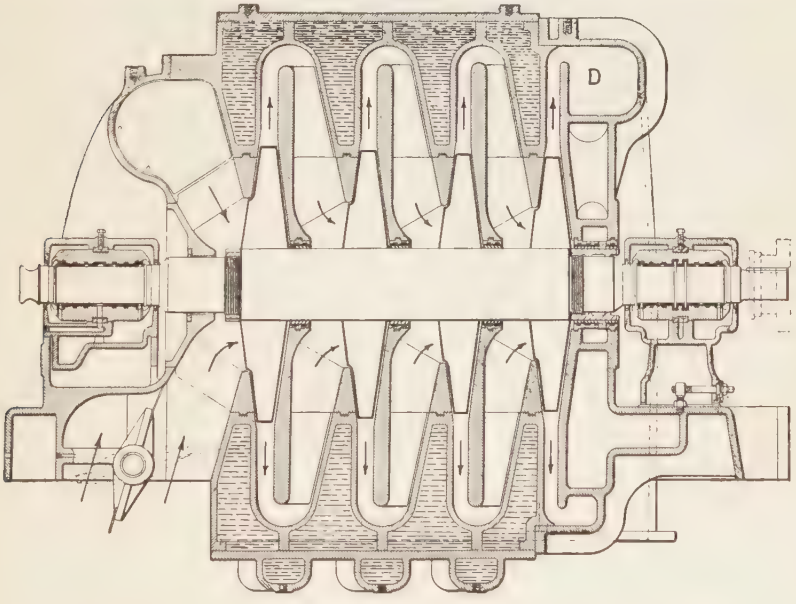


Fig. 252.—Section of an Elliott four-stage centrifugal air compressor.

four-stage centrifugal compressor. The rotor, Fig. 253, consists of straight radial impellers made of high-tensile-strength alloy steel; this

construction requires less weight of material than the shrouded type, similar to Fig. 246. The rapid rotation of the impellers (rarely less than 3000 r.p.m.) imparts to the air a velocity of 300 ft. per sec. or more. The velocity at which the air issues from each impeller is converted into pressure as the air passes into the larger passages leading to the next impeller, each adding an increment of pressure. Between the impellers are mounted curved guide vanes which direct the flow of air discharged by one impeller into the next stage at the proper angle in the direction of rotation. In multistage casings, water cooling is usually provided in the spaces between the impeller casings and passages, Fig. 252, for the partial removal of the heat of compression. A three-stage compressor of this type operating at 3600 r.p.m. can deliver 60,000 cu. ft. of air per

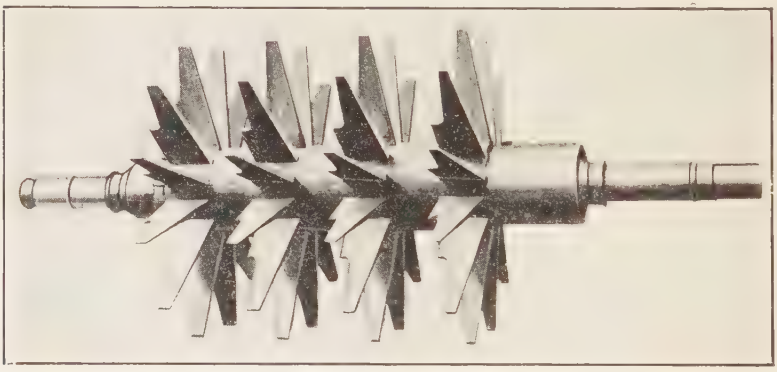


Fig. 253.—Rotor of an Elliott four-stage centrifugal air compressor.

minute against a pressure of 30 lb. per sq. in. gage. Centrifugal compressors are used for supplying compressed air for metallurgical service, exhausters, evaporators, grain elevators, pneumatic transmission, tools, sewage-disposal plants, sand blast, and gas boosters.

281. Piston Air Compressors.—The construction of the air cylinder of a piston compressor is, in general, similar to that of the cylinder in a steam engine. In a double-acting compressor each end of the cylinder must be fitted with an inlet and discharge valve corresponding to the inlet and exhaust valves of the steam cylinder. The valves may be actuated mechanically or by the air itself. The driving power for an air compressor is either a steam engine, electric motor, internal-combustion engine, or water wheel. The essentials of a piston air compressor are shown in Fig. 254; I being a spring-closed inlet valve and D the discharge valve. The indicator (or PV) diagram in Fig. 254 shows the changes of state of the air in its passage through the compressor.

Single-stage Compressor.—Air, Fig. 254, is drawn into the cylinder from the atmosphere along the line AB. The pressure in the cylinder

during the suction stroke is below that of the external or atmospheric pressure P_1. This can be accounted for by the fact that the pressure in the cylinder must be lower than that outside before the inlet valve will open. At B the piston starts on its return stroke and the inlet valve closes. If there were no heat loss due to radiation and no friction losses, the compression of the air would be adiabatic and would follow the line BC'. At point C' the discharge valve would open and the air would be forced into the receiver, the pressure following the line $C'D$. At D the discharge valve closes and the air in the clearance space expands from D to A as the piston starts on its return stroke. At A the inlet valve opens and the cycle starts again. If, by some means, the heat caused by compression could be taken away as fast as formed, the compression of the air would be isothermal and would follow the line BC''. The remainder of the cycle would be the same as before.

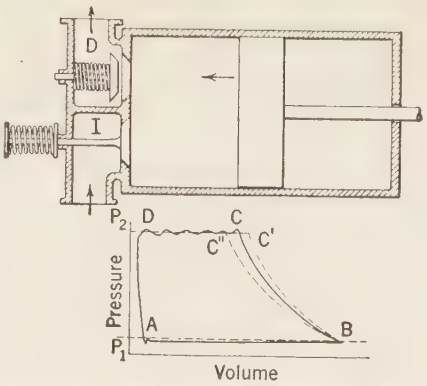


FIG. 254.—Diagrammatic piston air compressor and PV-diagram.

By reference to the PV diagram in Fig. 254 it can be seen that an equal weight of air can be compressed more economically by isothermal compression than by adiabatic compression. The area inside the figure $ABC'D$ represents the *work done on the air* during adiabatic compression. $ABC''D$ represents the work done on the air during isothermal compression, and area $C''BC'$ represents the saving effected by the use of isothermal instead of adiabatic compression. Efforts have been made to approach isothermal compression by: (1) injecting cold water as a spray into the air cylinder, (2) cooling the cylinder from without, that is, enveloping it in a cold water jacket, and (3) dividing the work into two or more steps or stages with an intercooler between successive stages. The use of either an atomized spray or a solid jet of cold water is effective as a means of cooling but introduces difficulties in the lubrication of cylinder and piston. In American practice the injection of water into the air cylinder has been abandoned, and it is generally customary to water-jacket piston air compressors. The air can be cooled partially with a water jacket and better cylinder lubrication obtained. The work done on the air by the *actual* single-stage compressor using a water jacket can be represented by the figure $ABCD$. It should be noted that the discharge line CD of this diagram lies above the receiver pressure line P_2. Discharge occurs along CD because the pressure inside

the cylinder must be greater than that outside before the discharge poppet valve will open and allow the air to escape.

For the adiabatic compression curve BC', Fig. 254, $PV^k = C$. Where $k = 1.40$ and for the isothermal compression curve BC'', $PV = C$. Because of the inability of the water jacket to absorb heat rapidly, compression occurs along the curve BC and $PV^n = C$, where the exponent n lies somewhere between 1 and 1.40. Under usual working conditions, n is between 1.28 and 1.35.

Multistage Compressor.—An effective method of approaching isothermal air compression (low value of n) is by multistage operation, the principle of which is illustrated by Fig. 255. Assume that A is the state at the beginning of compression, and that the path AB is practically adiabatic, even though jacket cooling is used. Let AF be an isothermal drawn through A. In multistage compression, the air follows path AB to some intermediate pressure P' at C. The air is discharged and cooled at constant pressure in an intercooler (external, water cooled) until the temperature of the air is approximately the same as at admission to the cylinder. CD represents this cooling. The air now passes to the second cylinder, where it is adiabatically compressed along DE and discharged along EF at pressure P_2 to the receiver. The diagram illustrates two-stage compression, but three or more stages are sometimes used. The work saved by the above two-stage compression over that of single-stage adiabatic compression is shown by the shaded area $CDEB$. If the final pressure exceeds 100 lb. per sq. in., the compression is more economically carried out in two or more stages. Multistage compression reduces the total work of compression. The stresses in the compressor are smaller than in a single-stage compressor of equal capacity and the temperature range in each cylinder is less than in a single cylinder. Thermodynamically, the efficiency of stage compression depends largely on the proper use of water jackets for the air cylinders, and the size and design of the intercoolers.

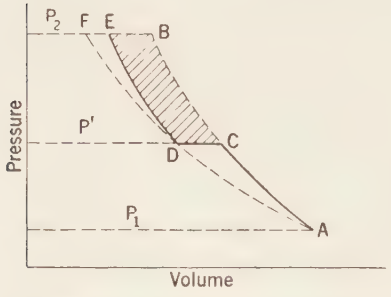


Fig. 255.—Pressure-volume diagram for multi-stage air compression.

282. Piston Air Compressor Classification and Details.—Reciprocating or piston air compressors may be classified according to the method of driving, number of stages, type of frame, method of cooling, kind of valves, etc. Steam-driven compressors are generally mounted as one unit with the steam cylinders and a common flywheel. Regulation is

usually effected by controlling the speed of the steam engine in accordance with the air pressure in the receiver; however, the steam-engine governor imposes a maximum speed limit. Compressors are also operated by electric motors, belt drives, internal-combustion engines, or water wheels. Constant-speed compressors may be regulated by means of an "unloading" valve, which either keeps the suction valve closed during one or more strokes or allows the air to discharge into the atmosphere when the desired receiver pressure is reached. One prominent manufacturer regulates the constant-speed compressor by a patented five-step clearance volume control which loads or unloads the compressor in five successive steps, viz., full load, three-quarters, one-half, one-quarter, and no load. With this clearance control a number of

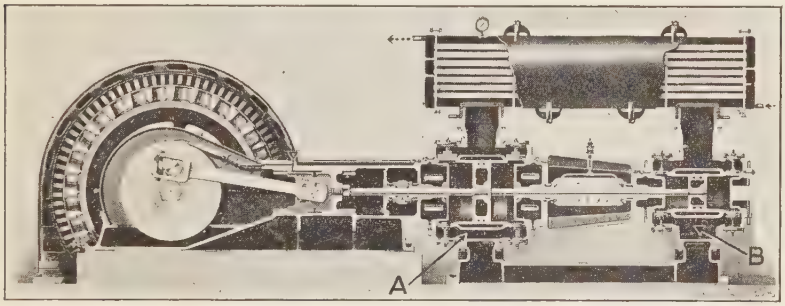


FIG. 256.—"Chicago Pneumatic" tandem compressor with overhead intercooler.

clearance pockets in the high- and low-pressure cylinders are thrown into communication with each cylinder during compression in proper sequence, the process being controlled by a predetermined air-receiver pressure. High-compression efficiencies are obtained at all loads, and the power required is in proportion to the load. In the variable-discharge type of compressor, the delivery may be regulated by closing the inlet valve before the completion of the forward (suction) stroke.

The ordinary small compressor is a single-stage reciprocating machine, with poppet air valves in the head or sides of the cylinder. Different arrangements of cylinders and pistons are possible in multistage compressors, the most common being the tandem and the cross-compound. Figure 256 shows a motor-driven two-stage double-acting compressor with the low-pressure and high-pressure cylinders in tandem with an intercooler between the stages. The inlet pipe connection to suction space *A* is not shown by Fig. 256. The pipe connection to discharge space *B* of the high-pressure cylinder has also been cut away. The intercooler consists of a cast-iron cylindrical shell, inside of which are straight brass tubes between the tube sheets. The air entering the

shell from the first stage surrounds the tubes and is compelled by baffles to cross the tube space several times on the way to the second stage. The cooling water is guided to the tubes by projecting ribs on the intercooler heads, which cause the water to pass from end to end several times. The design of the intercooler should be such as to insure efficient heat transfer without undue pressure loss. The pressure loss through the intercooler may be caused by shrinkage of air due to cooling or frictional resistance to the flow of air. The air valves in Fig. 256 are of the thin-plate spring-seated type, similar to Fig. 259. The tandem com-

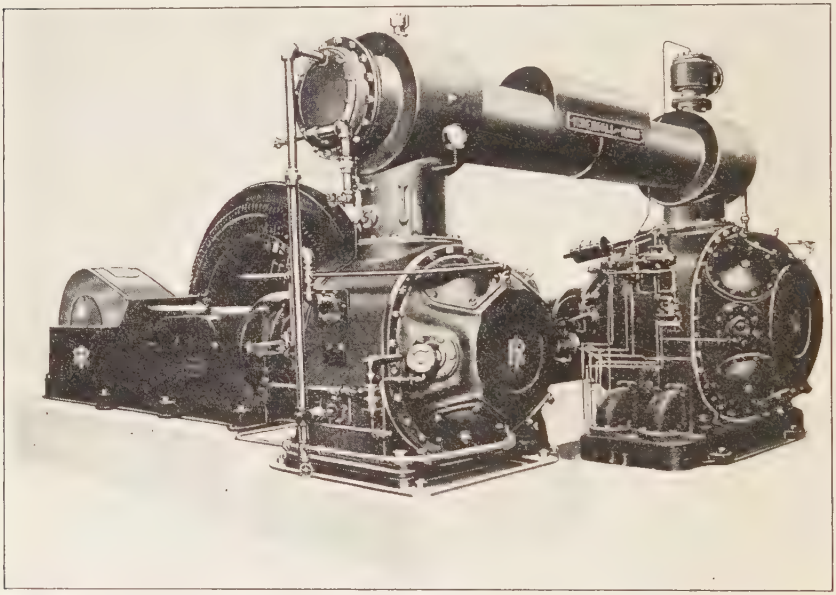


Fig. 257. Cross-compound compressor, electrically driven by a synchronous motor.

pressor is largely employed for small or medium-sized plants; the machine being compact, strong, and self-contained. Though useful for moderate air pressures and fairly constant loads, the steam-driven tandem-compressor is not capable of operating with good steam economy, nor is it self-regulating at less than 40 per cent of its rated load. These compressors are built up to capacities of 2000 cu. ft. of "free air"[1] per minute and 350 hp.

The cross-compound compressor, Figs. 257 and 258, is preferable to the tandem unit for continuous service in large plants. Standard machines of the cross-compound type are built in capacities up to 8000

[1] Cubic feet of free air is understood to be the actual amount of air drawn into the cylinder, measured at normal atmospheric conditions.

cu. ft. of free air per minute. A unit of this type is better adapted to varying loads, is self-regulating, and gives good power economy down to one-fourth rated load; also, its frictional losses are generally lower than those for the tandem type. However, the cross-compound installation requires more floor space and the first cost is higher than that of a tandem compressor of equal capacity.

Small reciprocating compressors are often driven by slide-valve steam engines. The Corliss type of steam engine is frequently used for the larger sizes, as the valve gear is well adapted for dealing with a

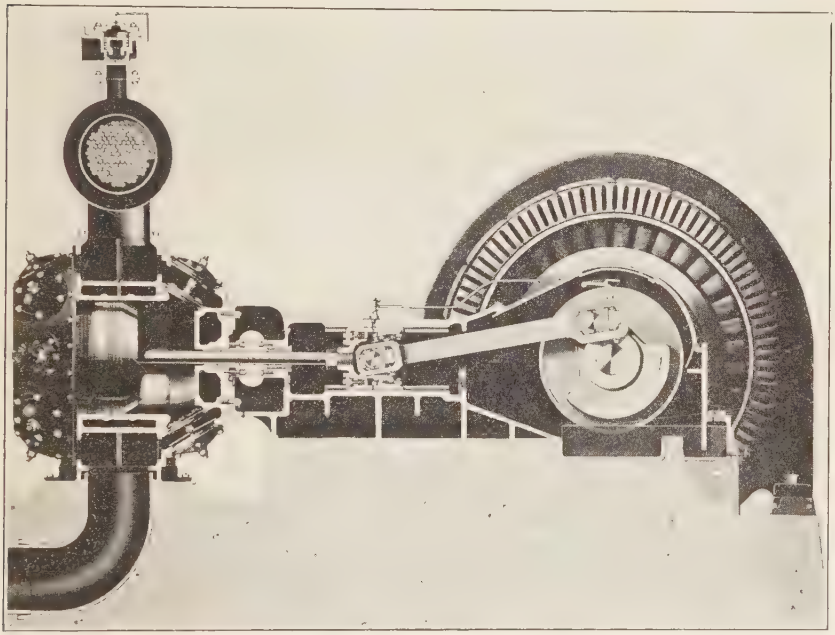
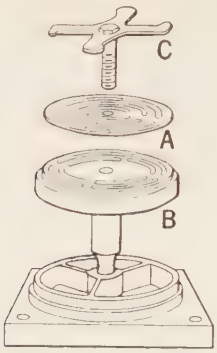


Fig. 258.—Section through an Ingersoll-Rand compressor cylinder.

variation of air pressure. Valves for air compression are generally of the automatic type as distinguished from the mechanically actuated type. For small air compressors, poppet valves are used extensively; for all other designs one of the various light-weight plate or strip valves may be used. These valves possess advantages of allowing the use of higher operating speeds, maximum area for a given diameter, quietness of operation, simplicity, durability, and sustained tightness. The valve assembly, Fig. 259, consists of the elastic valve plate *A* (a flat disk of special steel), the valve seat *B*, and the adjustable cover *C*; a number of small helical springs on the outer periphery separate *A* and *C*. On the suction stroke of the compressor, the plate is bowed upward

about $\frac{1}{8}$ in. and air enters the cylinder through the ports in the valve seat B. In closing, the plate stretches out again and is seated because of the helical springs and its own elasticity.

In most manufacturing plants, mines, quarries, etc., the presence of moisture in the compressed air is objectionable, on account of the difficulty of lubrication and the corrosion of air tools, the liability of freezing, and the rapid destruction of air-hose connections. Aftercoolers are employed to remove the moisture (and oil) present after the last stage of compression and to cool the air, thus decreasing the necessary size of the transmission pipe. Aftercoolers are similar in design and appearance to intercoolers. Intercoolers and aftercoolers are provided with a drain-cock for removing the moisture that collects. The receiver tank is also provided with a drain-cock, and has a safety valve and pressure gage.



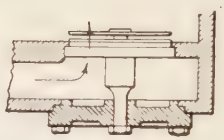


Fig. 259.—Plate-valve detail and assembly.

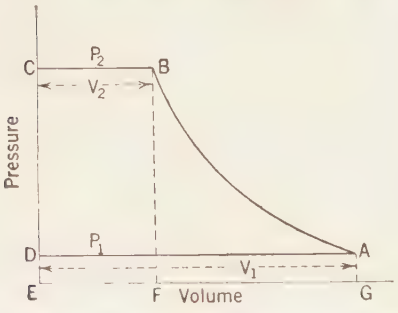


Fig. 260.—Theoretical PV-diagram for air compressor without clearance

283. The Ideal Reciprocating Air Compressor.—The area $ABCD$, Fig. 260, represents the net work delivered to the air by a compressor without clearance. Let P_1, V_1, and P_2, V_2, respectively, denote the pressures and volumes at points A and B. The work done on the air per cycle in foot-pounds is

$$W = {}_aW_b + {}_bW_c - {}_dW_a,$$

where ${}_aW_b$ = work of compression along AB, area $GABF$.

${}_bW_c$ = work of discharging the air from the cylinder, area $FBCDE$.

${}_dW_a$ = work done by atmospheric pressure on the piston during suction, area $DAGE$.

Assuming *adiabatic* compression, from equation 38, Chapter II,

$$_aW_b = \frac{P_2V_2 - P_1V_1}{k-1};$$

and

$$W = \frac{P_2V_2 - P_1V_1}{k-1} + P_2V_2 - P_1V_1,$$

or

$$W = \frac{k}{k-1}(P_2V_2 - P_1V_1); \tag{150}$$

but

$$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{k}}.$$

Substituting and reducing,

$$W = \frac{k}{k-1}P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right], \tag{151}$$

which is the general expression for the work of single-stage compressors with adiabatic compression, when the clearance is zero.

If the nature of the compression is such that PV^n is constant, n being any exponent, the work done per cycle would be

$$W = \frac{n}{n-1}P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]. \tag{152}$$

The above equation is the general expression for the work of single-stage compressors when the compression is "polytropic," according to the law $PV^n = C$, and when the clearance is zero.

For *isothermal* compression $PV = C$, from equation 41, Chapter II,

$$_aW_b = P_1V_1 \log_e\left(\frac{V_1}{V_2}\right).$$

Thus

$$W = P_1V_1 \log_e\left(\frac{V_1}{V_2}\right) + P_2V_2 - P_1V_1;$$

but

$$P_2V_2 = P_1V_1 \quad \text{and} \quad \frac{V_1}{V_2} = \frac{P_2}{P_1},$$

hence

$$W = P_1V_1 \log_e\left(\frac{P_2}{P_1}\right). \tag{153}$$

This is the general equation for the work of single-stage compressors operating isothermally, when the clearance is zero.

The volume of air at A, Fig. 260, which has been designated by symbol V_1 in the above equations, is generally taken as "free air" at intake temperature and pressure. By *free air is* meant *air at normal atmospheric conditions* where the compressor is installed. Because the altitude, barometer, and temperature vary at different localities, it follows that this term does not mean air under identical conditions. At sea level the normal atmospheric pressure is 14.7 lb. per sq. in. abs., at 3000 ft. above sea level 13.1 lb. per sq. in. abs., and at 8000 ft. above sea level 10.8 lb. per sq. in abs.

284. Net Work Done on the Air in Single-stage Compression.—An actual compressor cannot be operated without cylinder clearance, as there must always be a certain amount of mechanical clearance between the cylinder head and the dead-center position of the piston to insure safe operation. The clearance volume is somewhat increased by the valve passages between the valve faces and the inside of the cylinder. The air remaining in the clearance space expands on the return stroke of the piston, thus partially filling the cylinder and reducing the volume of external air taken in during the suction stroke. In Fig. 261, curve CD represents the "expansion" of the air in the clearance space. *The volume of the air entering the cylinder per stroke is represented by length* V_1, the piston displacement by V_p, and the clearance volume by V_c.

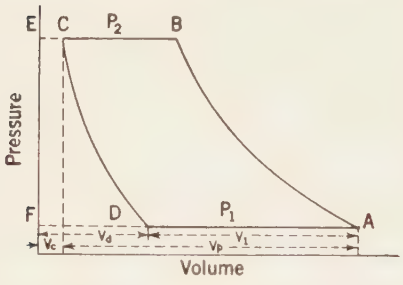


FIG. 261.—Theoretical PV-diagram for single-stage compression with clearance.

If the value of n for the curve CD is the same as for the compression AB, the net work done on the air per cycle is given by the same expression as in the case of the compressor without clearance, equation 152,

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right].$$

If curves AB and CD, Fig. 261, are adiabatic the work done on the air per cycle is equation 150,

$$W = \frac{k}{k-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right];$$

and if isothermal by equation 153,

$$W = P_1 V_1 \log_e \left(\frac{P_2}{P_1} \right).$$

The clearance does not affect the work done by the compressor; the air in the clearance space reacts like a spring, storing and giving back energy alternately. In other words, clearance affects the volumetric output of the compressor but not the work of compression for a given volume of air taken into the cylinder. The clearance of a compressor is the ratio of the clearance volume to the piston displacement, from Fig. 261, $\frac{V_c}{V_p} = m$, the clearance, and may vary from 1 to 5 per cent depending upon the size and design of the compressor.

The above work equations are sometimes more conveniently expressed in terms of cylinder displacement and "volumetric efficiency" than in the pressure-volume units. Referring to Fig. 261,

V_p = displacement volume of cylinder, cu. ft.

$m = \frac{V_c}{V_p}$ or $V_c = mV_p$,

$V = V_p(1 + m)$, total cylinder volume in cu. ft.

V_d = volume of re-expanded clearance air in cu. ft.

Then

$$V_d = V_c\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} = mV_p\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}};$$

but

$$V_1 = V_p + V_c - V_d.$$

Substituting,

$$V_1 = V_p + mV_p - mV_p\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}},$$

or

$$V_1 = V_p\left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]. \tag{154}$$

The ratio of the free air actually taken into a compressor cylinder to the volume displaced by the piston is called the *volumetric efficiency* of the compressor, or $e_v = \frac{V_1}{V_p}$. Substituting, in equation 152, the value of V_1 as obtained by equation 154,

$$W = \frac{n}{n-1} P_1V_p\left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]; \tag{155}$$

which expresses the work in terms of displacement and clearance, with pressures in pounds per square foot absolute.

But

$$e_v V_p = V_1 = V_p\left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right],$$

therefore

$$e_v = \left[1 + m - m\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right].$$

Substituting in equation 155, the expression for work done on the air reduces to

$$W = \frac{n}{n-1} e_v P_1 V_p\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]. \qquad (156)$$

285. Work of Multistage Compression.—Figure 262 shows the theoretical PV diagram for two-stage compression with clearance. The curve AB represents the compression in the first cylinder to the intermediate pressure P', line BE the cooling of the air at constant pressure to the initial temperature T_1, curve EG the compression from P' to P_2 in the second cylinder. The diagram shows a single continuous expansion line CD, which is taken to represent the expansion lines of both cylinders. On the PV plane, $ABFD$ is the indicator diagram for the first compressor and $EGCF$ the indicator diagram for the second. For a single-stage compressor, working between the same pressure limits P_1 and P_2, the indicator diagram would be $AHCD$; hence the saving of work by two-stage compression over single-stage compression is represented by the area $BEGH$.

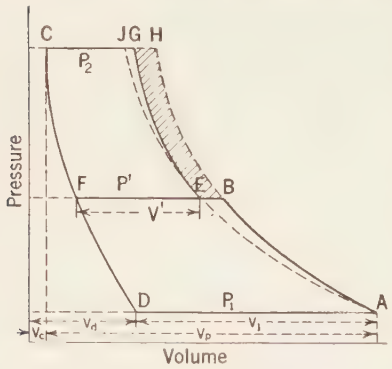


FIG. 262. Theoretical PV-diagram for two-stage compression with clearance.

According to equation 152, the work done in the first cylinder is given by the expression,

$$W_1 = \frac{n}{n-1} P_1 V_1\left[\left(\frac{P'}{P_1}\right)^{\frac{n-1}{n}} - 1\right].$$

The work done in the second cylinder is given by the expression

$$W_2 = \frac{n}{n-1} P' V'\left[\left(\frac{P_2}{P'}\right)^{\frac{n-1}{n}} - 1\right].$$

If the value of n for the expansion curve is the same as for the compression curve, and if point E is on the isothermal AEJ, then $P'V' = P_1V_1$ and therefore

$$W_2 = \frac{n}{n-1} P_1V_1\left[\left(\frac{P_2}{P'}\right)^{\frac{n-1}{n}} - 1\right].$$

Hence the general equation for the work of a two-stage compressor, with clearance and perfect intercooling, takes the form

$$W = W_1 + W_2 = \frac{n}{n-1} P_1V_1\left[\left(\frac{P'}{P_1}\right)^{\frac{n-1}{n}} + \left(\frac{P_2}{P'}\right)^{\frac{n-1}{n}} - 2\right]. \quad (157)$$

The pressures P_1 and P_2 are fixed; P' is variable. By differentiating the above equation with respect to the variable P' and making the first differential coefficient $\frac{dW}{dP'} = 0$, it is found that the maximum value is given when

$$P' = \sqrt{P_1P_2}, \quad (158)$$

or

$$\frac{P'}{P_1} = \frac{P_2}{P'} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{2}}. \quad (159)$$

This is the value for the best (most economical) intercooler receiver pressure.

If there are three stages with intermediate pressures P' and P'' the condition for minimum work of compression is

$$\frac{P'}{P_1} = \frac{P''}{P'} = \frac{P_2}{P''} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{3}}. \quad (160)$$

Combining equations 159 and 157, the total work of a two-stage compressor, with clearance and perfect intercooling and the intermediate pressure as given in (158), is,

$$W = \frac{2n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{2n}} - 1\right]. \quad (161)$$

Likewise, for three stages,

$$W = \frac{3n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{3n}} - 1\right]. \quad (162)$$

Example.—Find the cylinder dimensions and horsepower required to drive an air compressor at 200 r.p.m. that is to compress 1050 cu. ft. of air per minute to a pressure of 114 lb. per sq. in. abs. The exponent n for both compression and expansion will be assumed as 1.3 and the clearance ratio m as 0.03. Pres-

sure at the beginning of compression is 14 lb. per sq. in. abs. and inlet temperature 60 deg. F. Piston speed to be 600 ft. per min.

Solution.—*Single-stage Compressor.*

Displacement volume, equation 154,

$$V_p = \frac{1050}{1 + 0.03 - 0.03\left(\frac{114}{14}\right)^{\frac{1}{1.3}}} = 1193 \text{ cu. ft. per min.}$$

Displacement volume per cycle (double-acting),

$$V'_p = \frac{1193}{2 \times 200} = 2.98 \text{ cu. ft.}$$

Stroke, $$l = \frac{600}{2 \times 200} = 1.5 \text{ ft.} = 18 \text{ in.}$$

Effective piston area $= \frac{2.98}{1.5} \times 144 = 286.1$ sq. in.

Allowing 3.1 sq. in. for a 2-in. piston rod, the gross area should be 289.2 sq. in., and the corresponding diameter is $19\frac{1}{4}$ in.

The work per minute, from equation 152, is

$$W = \frac{1.3}{0.3} \times 144 \times 14 \times 1050\left[\left(\frac{114}{14}\right)^{\frac{0.3}{1.3}} - 1\right]$$

$$= 5{,}687{,}200 \text{ ft-lb. per min.}$$

Therefore the power developed by the compressor $= \frac{5{,}687{,}200}{33{,}000} = 172.3$ air hp. If there is a 15 per cent loss between the steam engine and compressor the horsepower developed in the steam engine should be

$$172.3 \div 0.85 = 202.7 \text{ steam hp.}$$

Two-stage Compressor.

Intermediate pressure $P' = \sqrt{P_1 P_2}$,

$$P' = \sqrt{14 \times 114} = 39.96 \text{ lb. per sq. in. abs.}$$

Assuming a pressure drop through the intercooler as one pound, take 40.5 lb. as the discharge pressure of the low-pressure cylinder and 39.5 lb. as the intake pressure to the high-pressure cylinder.

Low-pressure cylinder displacement volume,

$$V_p = \frac{1050}{1 + 0.03 - 0.03\left(\frac{40.5}{14}\right)^{\frac{1}{1.3}}} = 1092 \text{ cu. ft. per min.}$$

Using a stroke of 18 in., effective piston area

$$= \frac{1092}{2 \times 200 \times 1.5} \times 144 = 262 \text{ sq. in.}$$

Allowing 3 sq. in. for the piston rod, the corresponding low-pressure cylinder diameter is $18\frac{1}{2}$ in.

The air passing through the intercooler has its temperature reduced to the initial temperature of the air entering the low-pressure cylinder. Hence the ratio 14 to 39.5 is the relation between the air volumes of the high-pressure cylinder and the same volumes for the low-pressure cylinder.

High-pressure cylinder displacement volume $= 1092 \times \frac{14}{39.5} = 387$ cu. ft. per min.

Using a stroke of 18 in., effective piston area $= \frac{387}{2 \times 200 \times 1.5} \times 144$ $= 93$ sq. in.

Allowing 3 sq. in. for the piston rod, the corresponding high-pressure cylinder diameter is 11 in.

The work of the low-pressure cylinder is, from equation 152,

$$W_1 = \frac{1.3}{0.3} \times 144 \times 14 \times 1050 \left[\left(\frac{40.5}{14} \right)^{\frac{0.3}{1.3}} - 1 \right];$$

$$W_1 = 2{,}548{,}200 \text{ ft-lb. per min.}$$

Power developed in low-pressure cylinder $= \frac{2{,}548{,}200}{33{,}000} = 77.22$ air hp.

The work of the high-pressure cylinder is, from equation 152,

$$W_2 = \frac{1.3}{0.3} \times 144 \times 14 \times 1050 \left[\left(\frac{114}{39.5} \right)^{\frac{0.3}{1.3}} - 1 \right];$$

$$W_2 = 2{,}541{,}800 \text{ ft-lb. per min.}$$

Power developed in the high-pressure cylinder $= \frac{2{,}541{,}800}{33{,}000} = 77.02$ air hp.

Total air horsepower required for two-stage compression = 154.24, which is about 10 per cent less than was required for the single-stage compressor.

286. Compressor Indicator Diagrams.—The actual indicator diagram departs considerably from the ideal, the real compressor requiring a greater expenditure of work on the air than the ideal machine because of clearance, leakage, friction, inefficient cooling, imperfect valve action, etc. Typical high- and low-pressure diagrams for two-stage compression are shown in Fig. 263. The indicator cards have been reduced to the same scale of volumes and pressures; the figure is called *the combined card*. The indicator card from each cylinder was divided into an equal

number of parts and the pressures and volumes at each of these points were computed and plotted to obtain Fig. 263. Starting at D, the point of opening of the low-pressure suction valve, it should be noted that pressure P_1 is below atmospheric pressure. This is due to the fact that the internal pressure must be somewhat lower than the external pressure before the poppet valves will open or because of restricted inlet valve area. At A the suction valves close and the air is compressed along line AB. The adiabatic line AH falls to the right of AB, and the saving due to water jacketing the cylinder is represented by area BAH. The discharge pressure in the low-pressure cylinder is greater than the intercooler pressure P', owing to the friction and sticking of the discharge valves. Discharge in the low-pressure cylinder ceases at F, and the clearance air expands from F to D before the inlet valve will again open. The wavy suction and discharge lines are caused by oscillations of the air column and oscillations or "fluttering" of the valves. AC is the

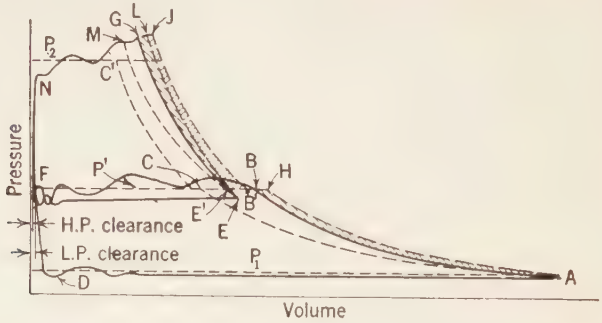


Fig. 263.—Combined (indicator) card for two-stage compression.

isothermal compression line for the low-pressure cylinder. The area CAH represents the work that could have been saved if perfect cooling had taken place.

If there were no difference in the clearance volumes of the two cylinders and the expansion lines crossed the intercooler pressure line at the same point, then, with no cooling in the intercooler, the compression in the high-pressure cylinder would be continued from point B, Fig. 263. Assuming no cooling in the intercooler the actual high-pressure cylinder compression would start at B' instead of B, owing to the difference in clearance volumes in the two cylinders, and adiabatic compression would take place along $B'J$. In the actual case the volume of air is decreased by cooling in the intercooler, and the high-pressure compression starts at E instead of B'. Because of friction losses through ports and valves, the actual suction line of the high-pressure cylinder is below the intercooler pressure P'. The actual compression line in the high-pressure cylinder is shown by line EG, and the work saved due to high-

pressure cylinder jacketing is shown by area *GEL*. Area *B'E'LJ* represents the appreciable saving in work done due to intercooling. The isothermal compression line for the high-pressure cylinder is *EM* and the theoretical saving possible due to cooling in the high-pressure cylinder is represented by area *MEL*. The discharge valve does not open until the pressure attained in the high-pressure cylinder is above that of receiver pressure P_2. There are generally several oscillations of the valve and air column, and the discharge line generally obtained is somewhat like *JMN*. The discharge valves on some compressors do not close suddenly at the end of the stroke, consequently the corner at *N* may be more or less rounded.

The overlapping of the high- and low-pressure cards, Fig. 263, indicates a loss, because the work represented by this area is in reality work done twice. The drop in pressure between the cylinders is caused by the resistance of the discharge valves of the low-pressure cylinder, the resistance of the inlet valves of the high-pressure cylinder, together with the friction in the air passages and intercooler, and the reduction of air volume produced by the temperature drop in the intercooler. In well-designed compressors this lost work is small because of the use of liberal valve areas, positively operated valves, and large and properly designed intercoolers with direct and ample passages to the cylinders. If the excess work expended on the air represented by this overlapping area exceeds the saving effected by intercooling (area *JB'E'L*), no economic benefit is derived from the use of the intercooler. There may also be an unequal distribution of the work in the two cylinders, and the two-stage compression may not realize a saving of power over single-stage compression for the same pressure range of P_1 to P_2.

287. Performance of Air Compressors.—The *displacement* of a compressor is the volume displaced or swept through by the net area (piston area less allowance for piston rod) of the compressor piston during the compression stroke. Piston displacement is usually expressed in cubic feet per minute. For multistage compressors, the low-pressure cylinder only is considered.

The *capacity* of a compressor is the actual volume of air compressed and delivered, usually expressed in cubic feet per minute at intake temperature and pressure. The capacity can be determined accurately only by actual measurement of the air output. The *volumetric efficiency* (ratio of the capacity, as defined above, to the displacement of a compressor) is decreased by excessive clearance, suction friction, heating during suction, and installation at high altitudes. The volumetric efficiency may be increased by reducing the temperature of the entering air, for instance, taking air from outdoors instead of from inside of a building.

Compression efficiency is the ratio of the isothermal horsepower required to compress the amount of air delivered, to the actual indicated horsepower required in the air-compressing cylinders as shown by indicator diagrams. The *mechanical efficiency* of a compressor is the ratio of the total i.hp. of the air cylinders to the steam i.hp., in the case of a steam-driven unit. For a power-driven machine the mechanical efficiency is the ratio of the air i.hp. to the brake hp. of the power drive (turbine, motor, or belt). The *overall* (sometimes called *compressor*) *efficiency* is the product of the compression efficiency and the mechanical efficiency, and represents the ratio of the isothermal horsepower of the compressor to the horsepower supplied by the power drive.

288. The Air Engine.—An engine using compressed air as a medium does not differ essentially from a steam engine; in fact, a steam engine may be used without change. Air engines are often used in mines, quarries, on locomotives, tunneling, for driving small portable tools, and in other applications. For small power units, a simple form of rotary engine is sometimes employed on account of convenience irrespective of low efficiency. The absence of heat, the ease of transmission over great distances without excessive pressure loss, and the absence of odor, obnoxious vapors, or fumes make air a popular medium where a number of small engines are to be operated at widely scattered points. The exhausted air aids in ventilating the rooms or mine passages in which it is used. Steam, on the other hand, cannot be efficiently transmitted through long pipelines because of the resulting condensation. Small steam engines are very inefficient, and the high temperature at which they operate renders them unsuitable where handling is required.

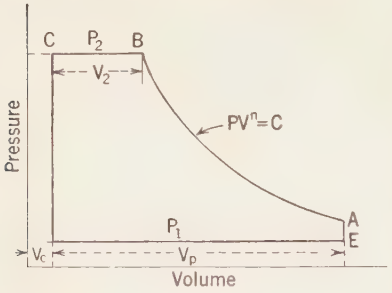


FIG. 264.—Theoretical air engine indicator diagram.

If the expansion is complete, the cycle of the air engine is that of the compressor traversed in the opposite direction, and the power delivered by the engine may be calculated by the equation for the power of the compressor; see equations 152 and 156. If the expansion is not complete, the net work done as shown by the indicator diagram can be found by integrating the areas under the individual curves. Thus, if Fig. 264 represents the indicator diagram of an air engine, which has clearance V_c and operates with incomplete expansion as shown by AE, the net work done *by* the air per cycle is

$$W = P_2(V_2 + V_c) + \frac{P_2(V_2 + V_c) - P_a(V_p + V_c)}{n - 1} - P_1V_p - P_2V_c.$$

Therefore

$$W = P_2V_2 - P_1V_p + \frac{P_2(V_2 + V_c) - P_a(V_p + V_c)}{n - 1}; \qquad (163)$$

where P_a is the absolute pressure at the point of release, A in Fig. 264.

Example.—Find the theoretical horsepower that can be developed by a double-acting 7-in. by 10-in. hoisting engine operating on compressed air and running at 125 r.p.m. Also estimate the air consumption in cubic feet per i.hp. per hour at 14.7 lb. abs. and 60 deg. F. The clearance is 3 per cent; temperature of inlet air, 80 deg. F., and pressure 75 lb. per sq. in. abs.; release pressure 39.2 lb. abs., and exhaust 14.5 lb. abs. Cut-off at 60 per cent of the stroke, and $PV^{1.32} = C$ during expansion.

Solution.—Substituting in (163),

$$W = 144\left[(75 \times 0.60D) - (14.5 \times 1.0D) + \frac{75(0.60D + 0.03D) - 39.2(1.0D + 0.03D)}{1.32 - 1.00}\right]$$

$$= 144[45D - 14.5D + 21.5D] = 7490D \text{ ft-lb.}$$

$$D = \frac{\pi \times 49 \times 10}{4 \times 144 \times 12} = 0.223 \text{ cu. ft. per stroke.}$$

$$\text{(Neglecting area of piston rod) i.hp.} = \frac{7490 \times 0.223 \times 2 \times 125}{33{,}000} = 12.64.$$

Estimation of Air Consumption.—At point B, the total volume of air is $0.63D$. But $0.03D$ is clearance air, used repeatedly and not chargeable to the engine. However, the $0.03D$ air at 14.5 lb. abs. reduces to $0.0058D$ at 75 lb. abs., and the volume of air brought into the cylinder per stroke at 75 lb. abs. is $0.6242D$ and the volume of air per hour is

$$0.6242 \times 0.223 \times 2 \times 125 \times 60 = 2088 \text{ cu. ft.}$$

$2088 \times \frac{75 \times 520}{14.7 \times 540} = 10{,}260$ cu. ft. per hour or 812 cu. ft. of air at 14.7 lb. abs. and 60 deg. F. per i.hp. per hour.

If the air is not preheated before entering the engine cylinder, the temperature at the end of expansion is necessarily low. Thus if air enters at 80 deg. F. and expands, according to $PV^{1.32} = C$, from 75 to 25 lb. per sq. in. abs., the final temperature is

$$T_a = (80 + 460)\left(\frac{25}{75}\right)^{\frac{1.32-1}{1.32}} = 414 \text{ deg. abs.,}$$

hence

$$t_a = -46 \text{ deg. F.}$$

The fall of temperature during expansion produces condensation of water vapor present in the air and may lead to the formation of ice in

the exhaust valves and passages of the engine or air motor. This difficulty may be counteracted in various ways: by circulating warm air or water around the exhaust passages, by specially designed exhaust ports, by a reduced range of pressures, by avoiding adiabatic and approaching isothermal expansion, and by thoroughly drying the air. The only commercially successful method of avoiding inconveniently low temperatures after expansion is by raising the temperature of the inlet air. Methods of reheating are: (1) Passing the air to be heated through a coil of pipe or cast iron exposed to a fire, hot gases, or steam; (2) the injection of steam or hot water into the air itself; and (3) placing an electric heating coil in the air line. The first method of heating is the most efficient and preferable from a mechanical standpoint.

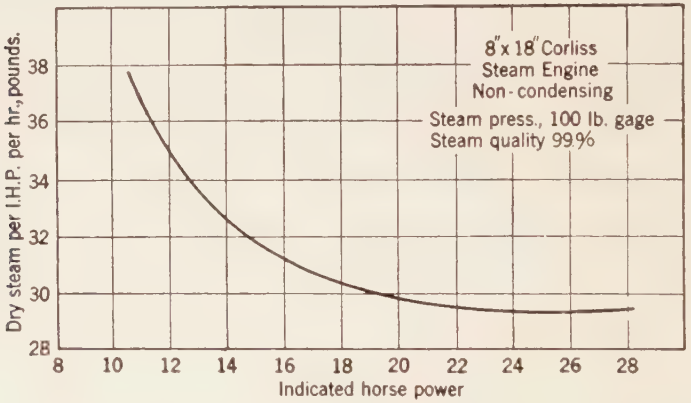


Fig. 265.—Steam consumption for 25-horsepower steam engine.

Small engines, operating without expansion, have an air consumption of about 1500 to 2200 cu. ft. (115 to 170 lb.) of air (measured at 14.7 lb. abs. and 60 deg. F.) per hp. per hr. Using the air expansively in these engines reduces the air consumption to a range of 700 to 1300 cu. ft. (50 to 100 lb.), depending somewhat upon the air pressure, point of cut-off, and horsepower of the engine.

Figures 265 and 266 show the results of tests [2] using steam and air, respectively, as working mediums in an 8-in. by 18-in., 25-hp. Corliss steam engine with pressures and temperatures as noted. The curves in Fig. 266 show that the economy of an air engine is dependent upon the amount of preheating; thus with the initial temperature at 200 deg. F. the economy is 42 lb. of air per i.hp. per hr. at 25 hp. That quantities of mixed air and steam are capable of doing more work expansively

[2] "A Study of Air-Steam Mixtures," by L. A. Wilson and C. R. Richards; Bulletin 131, 1922, University of Illinois Experiment Station.

than equal quantities of the two media acting separately can accomplish, is evidenced by further tests on the above engine. The results show that the maximum economy occurs with mixtures containing 30 per cent steam (by weight).

289. Other Uses of Compressed Air.—Aside from the driving of engines, high-pressure air is being used for a variety of purposes because of the convenience and ease of transmitting air as compared with steam. Some of the important applications of compressed air are: rock drills, hammer drills, chipping, calking and scaling hammers, riveters, air hoists, lifts and jacks, sand-blast nozzles for cleaning castings and removing paint, pneumatic painting, cement guns, cleaning boiler tubes, agitating liquids, pneumatic conveying systems, for pumping liquids

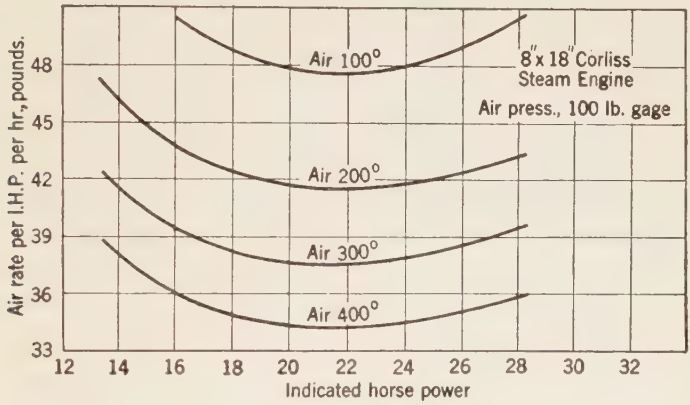


FIG. 266.—Air consumption for 25-horsepower engine.

by direct-acting pumps, by pneumatic displacement pumps, or by the ingenious air-lift system. Extensive industrial applications of compressed air have resulted in the perfecting of modern interchangeable pneumatic tools. The gage pressure usually required for the economical operation of pneumatic tools is 70 to 90 lb. per sq. in., the pressure depending somewhat upon the type of service.

The decrease of temperature accompanying a reduction of pressure is utilized in the manufacture of liquid air. Air is compressed to about 2000 lb. pressure in a three-stage machine and discharged to a cooler. The air is next delivered to the inner tube of a long (350 ft.) double coil, through which it passes to a small orifice called an expansion valve, where the air expands into a reservoir (maintained at about 325 lb. per sq. in. gage pressure). The throttling in the expansion valve causes a drop in temperature, and the cooled and expanded air returns through the outer tube of the coil back to the compressor, cooling the descending air in the inner pipe. At each passage during this regenerative process, a decrease

of temperature occurs; the effect is cumulative, and ultimately the temperature in the reservoir becomes lower than the critical temperature of air (−220 deg. F.), and liquefaction begins.

The drop in temperature during expansion in an air engine cylinder, and the difficulties which it occasions, were discussed in Art. 288. This low-temperature exhaust air may be utilized for refrigeration by being forced through pipes to a cold room for the cooling of its contents, the production of ice, etc. In some cases, the refrigerative effect alone may be sought, the performance of work during expansion being incidental.

290. Measurement of Air Flow. —The quantity of air delivered by a blower or air compressor may be determined by one of the following methods: pitot tube, Venturi tube, thin-plate orifice, rounded orifice, short tube or nozzle, measuring tanks of known capacity, causing air to displace water in a tank, and recording meters. The results obtained are usually reduced to terms of volume at the temperature and pressure existing at the compressor or blower intake. The method employing the pitot tube has been described in Chapter X. The Venturi tube has been explained in Art. 278 and illustrated by Fig. 249.

For determining the quantity of air flowing through a Venturi tube, when $p_2 > 0.53p_1$, G. A. Goodenough [3] gives

$$W = 2.056\frac{A_2 p_1}{\sqrt{T_1}}\sqrt{\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}}; \qquad (164)$$

where W = weight of air flowing, lb. per sec.
A_2 = area of throat, sq. in.
p_1 = entrance pressure, lb. per sq. in. absolute.
p_2 = throat pressure, lb. per sq. in. absolute.
T_1 = absolute temperature of air at entrance.
k = ratio of specific heats, $\frac{c_p}{c_v} = 1.4$ for air.

Example.—The inlet diameter of a Venturi tube is $2\frac{1}{2}$ in. and the throat diameter is $\frac{3}{4}$ in. Air flowing through the tube has an inlet temperature of 165 deg. F., an inlet pressure of 54 lb. per sq. in abs., and a throat pressure of 30 lb. per sq. in. abs. How many pounds of air are flowing per minute?

Solution.—

$$\frac{p_2}{p_1} = \frac{30}{54} = 0.556$$

$$W = 2.056\frac{0.4418 \times 54}{\sqrt{625}}\sqrt{(0.556)^{1.43} - (0.556)^{1.714}}$$

$$= 0.50 \text{ lb. per sec.} = 30 \text{ lb. per min.}$$

[3] "Principles of Thermodynamics," G. A. Goodenough, Chapter IX, Third Edition, 1920; Holt and Company.

The *thin-plate orifice* in the simplest and most familiar form is a round hole in a thin, flat diaphragm (about $\frac{1}{16}$ in. thick) clamped between the flanges at a joint in a pipeline, the hole being concentric with the pipe; see Fig. 267. A thicker plate is sometimes used, and the plate is then chamfered around the hole on the downstream side, while the entrance side of the plate remains flat, with a sharp 90-deg. corner at the edge of the opening. Pressure connections are provided in the pipe walls on both sides of the diaphragm, so that pressure gages or manometers may be attached and the decrease of pressure through the orifice may be determined. This difference in pressure is utilized to find the rate of flow. The thin-plate orifice employs the Venturi principle, the orifice merely acting as a means whereby the fluid forms its own Venturi shape beyond the orifice plate. When the pressure connections are properly located the Venturi equation would apply, if the diameter of the minimum section (or vena contracta) of the air jet could be measured directly. Since the diameter of the vena contracta is not directly measurable it must, in practice, be replaced by the diameter of the orifice which can be measured. Experiments have shown that the

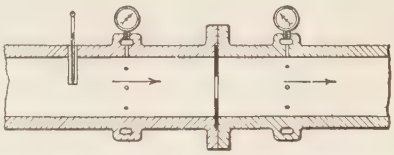


Fig. 267. –Thin-plate orifice installation.

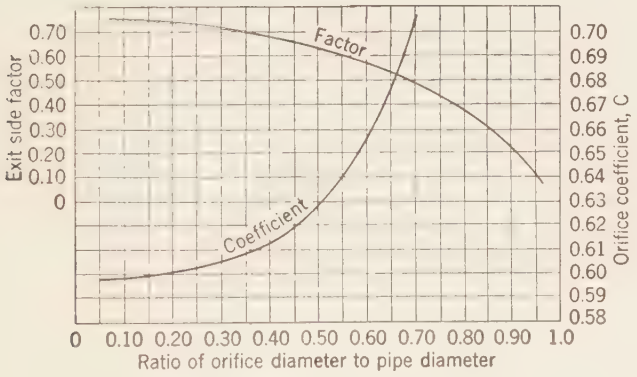


Fig. 268.—Constants for thin-plate orifices.

position of minimum pressure, at the vena contracta, is geometrically similar in 3-in. to 12-in. pipes for the same orifice ratios. The pressure connection on the inlet side of the orifice is usually located one pipe diameter from the orifice. The distance from the orifice to the pressure connection, at the middle of the vena contracta, on the exit side may be

determined by multiplying the pipe diameter by factors given by E. G. Bailey,[4] and shown in Fig. 268.

The rate of flow through a thin-plate orifice is

$$V = C\sqrt{2gH_a};$$

in which V = velocity, ft. per sec.
C = orifice coefficient, see Fig. 268.
g = acceleration due to gravity, 32.17 ft. per sec.^{2}
H_a = differential pressure, ft. of air.

But

$$Q = \frac{AV}{144}, \text{ cu. ft. per sec.,}$$

where A = area of orifice sq. in.

Hence

$$Q = \frac{\pi D^2}{144 \times 4} C\sqrt{64.34H_a} = 0.0438CD^2\sqrt{H_a}.$$

However,

$$H_a = \frac{h_w \times d_w}{12 \times d_a}.$$

Here h_w is the differential pressure across the orifice in inches of water, d_w and d_a are the densities in pounds per cubic foot of water in the manometer and of the air flowing, respectively.

Therefore

$$Q = 0.0438CD^2\sqrt{\frac{d_w \times h_w}{d_a \times 12}},$$

and

$$Q = 0.0126CD^2\sqrt{h_w \frac{d_w}{d_a}},$$

but

$$h_w = \frac{1728p}{d_w};$$

thus

$$Q = 0.0126CD^2\sqrt{\frac{1728pd_w}{d_a d_w}},$$

$$Q = 0.526CD^2\sqrt{\frac{p}{d_a}}; \qquad (165)$$

[4] "Fluid Meters, Their Theory and Application," Part I, Third Edition, 1931; A.S.M.E. Research Publication.

in which Q = air flowing through orifice, cu. ft. per sec. at exit pressure and entrance temperature.
C = orifice coefficient from Fig. 268.
D = diameter of orifice, in.
p = observed differential pressure across orifice, lb. per sq. in.
d_a = density of air, lb. per cu. ft. at exit pressure and entrance temperature, from $PV = MRT$, or Table XXIV.

The orifice coefficient varies according to the ratio of the orifice diameter to pipe diameter as shown in Fig. 268. This coefficient represents the product of two factors, a frictional coefficient which allows for the frictional resistance at the entrance to the orifice, and the contraction of the stream flow which is the ratio of the area of the vena contracta to the area of the orifice.

R. J. Durley in Vol. 27 of *Transactions* of the A.S.M.E., under "Air Flowing into the Atmosphere through Circular Orifices," discusses air flow through a thin-plate orifice when discharging directly to the atmosphere. The form of the equations given is similar to those stated above.

Example.—Air is flowing through a $\frac{3}{4}$-in. thin-plate orifice, the latter being inserted between the flanges of a 3-in. pipe. The pressure on the upstream side (3 in. from orifice) is 20 in. of mercury above atmosphere, and the pressure on the downstream side (at the vena contracta, $2\frac{3}{16}$ in. from orifice) is 4 in. of mercury. Find the cubic feet of air flowing per minute, if the temperature of the air on the upstream side is 140 deg. F. and the barometer is 29.32 in. mercury.

Solution.—

$$p = (20 - 4) \times 0.49 = 7.84 \text{ lb. per sq. in.}$$

$$C = 0.603, \text{ from curve, Fig. 268.}$$

$$d_a = \frac{M}{V} = \frac{P}{RT} = \frac{(4 + 29.32) \times 0.49 \times 144}{53.35 \times 600}.$$

$$d_a = 0.0735 \text{ lb. per cu. ft.}$$

Substituting in equation 165,

$$Q = 0.526 \times 0.603 \times \overline{0.75}^2 \sqrt{\frac{7.84}{0.0735}},$$

$$Q = 0.18 \times 10.34 = 1.845 \text{ cu. ft. per sec.}$$

$$= 110.7 \text{ cu. ft. per min. at 4 in. of mercury and 140 deg. F.}$$

The low-pressure nozzle test is recommended by the Compressed Air Society as an accurate method for measuring the air output of a compressor. The air delivered by the compressor is discharged into receiver A, Fig. 269, where the desired pressure is maintained constant by bleeding air through globe valve V to a second receiver B provided with a nozzle N. The diameter of B should be at least two and a half ($2\frac{1}{2}$) times the smallest nozzle diameter, preferably larger; B should be 10 to 15 or more feet long. This nozzle is merely an orifice open to the atmosphere with curved inner edges and is of such size that the pressures in receiver B will not be more than one pound per square inch above atmosphere, when passing all the air through its opening. The exact pressure in B is determined by a water manometer M, and the temperature of the air flowing is measured by a thermometer T (no well) passing through the wall of receiver B, about 2 ft. from the nozzle. From these observations, together with the barometric reading, the cubic feet of air flowing through the nozzle per second may be determined. Figure 270 shows a drawing of the low-pressure nozzle with an accompanying

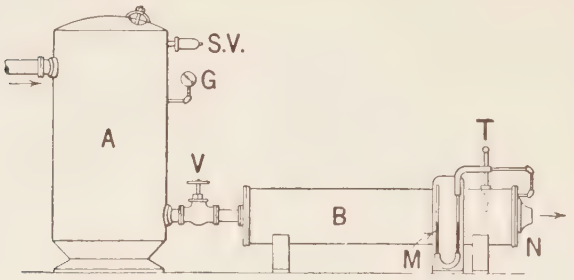
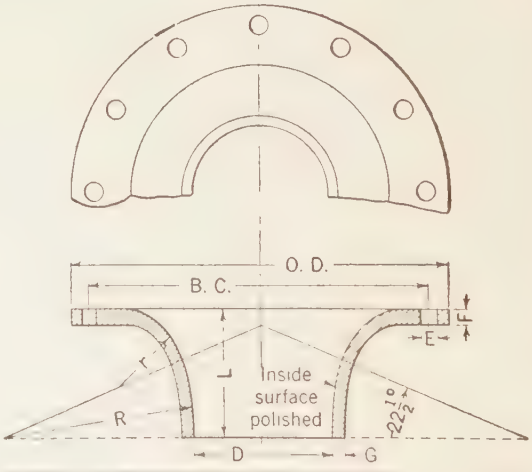


Fig. 269.—Arrangement of apparatus for air measurement by low-pressure nozzle.



D	L	R	r	E	I O. D.	I B C.	No. of bolts	Drilled for std. flange size	Free air cu. ft. max. capacity	Free air cu. ft. min. capacity	F	G
2"	1.70"	2.8"	1.0"	⅞"	15·0"	13¼"	12	9"	340	140	⅝"	½"
3"	2.55"	4.2"	1.5"	⅞"	15·0"	13¼"	12	9"	755	290	⅝"	½"
4"	3.40"	5.6"	2.0"	⅞"	15·0"	13¼"	12	9"	1330	520	⅝"	½"
5"	4.25"	7.0"	2.5"	⅞"	15·0"	13¼"	12	9"	2065	810	⅝"	½"
7"	5.95"	9.8"	3.5"	1"	19·0"	17·0"	12	12"	3990	1560	¾"	½"
9"	7.65"	12.6"	4.5"	1⅛"	23½"	21¼"	16	16"	6600	2575	¾"	½"
11"	9.35"	15.4"	5.5"	1¼"	29½"	27¼"	20	22"	9860	3850	¾"	½"

Fig. 270.—Low-pressure air nozzle details, Compressed Air Society.

table of dimensions and capacities as recommended by the Compressed Air Society. The inner surface of the nozzle must be highly polished, giving a nozzle coefficient of 0.98 to 0.99, depending upon the shape and finish.

When the nozzle, as selected by the table given in Fig. 270, is of such size that the manometer reading is between 3 and 20 in. of water, the quantity of air flowing through the nozzle may be calculated by equation 165, but using a new nozzle coefficient, thus

$$Q = 0.526KD^2\sqrt{\frac{p}{d_a}}; \tag{166}$$

where Q = cubic feet of air flowing per second at observed temperature of upstream side of nozzle and pressure of downstream side (usually atmospheric).

K = nozzle coefficient, 0.98 to 0.99.

D = diameter of smallest part of nozzle throat, in.

p = observed differential pressure across nozzle, lb. per sq. in.

d_a = density of air in pounds per cubic foot at exit pressure and entrance temperature, from $PV = MRT$.

Example.—The air discharged from an air compressor flows through a 4-in. Compressed Air Society low-pressure nozzle into the atmosphere. The observed manometer reading is 12 in., the temperature of the air entering the nozzle is 206 deg. F., and the barometer is 29.40 in. of mercury. How many cubic feet of air are being compressed per minute if the nozzle is of such shape and finish that its coefficient is 0.99? Temperature of the liquid in the manometer is 80 deg. F.

Solution.—

$$\text{In equation 166, } p = 12 \times \frac{62.22}{1728} = 0.432 \text{ lb. per sq. in.}$$

$$d_a = \frac{M}{V} = \frac{P}{RT} = \frac{29.4 \times 0.49 \times 144}{53.35 \times 666},$$

$$d_a = 0.0584 \text{ lb. per cu. ft.}$$

Hence

$$Q = 0.526 \times 0.99 \times 4^2\sqrt{\frac{0.432}{0.0584}},$$

Q = 22.7 cu. ft. per sec.

= 1362 cu. ft. per min. at 206 deg. F. and atmospheric pressure.

PROBLEMS

1. The discharge pressure of a single-stage air compressor was 100 lb. per sq. in. abs. Inlet air temperature, 70 deg. F.; suction pressure, 14.3 lb. per sq. in. abs.; inlet air handled, 376 cu. ft. per min. If the compression was according to $PV^{1.3} = C$, find the discharge temperature in degrees Fahrenheit.

2. What was the percentage of excess power required for the compressor of problem 1 as compared with isothermal compression? Assume the cycle to be like *ABCD*, Fig. 260.

3. How many horsepower was saved by the water jacket in problem 1, as compared with adiabatic compression?

4. Find the horsepower required to compress the air of problem 1 if it were first compressed to a pressure of 38 lb. per sq. in. abs., cooled to the initial temperature at constant pressure, and then compressed to a pressure of 100 lb. per sq. in. abs.

5. Determine the horsepower required to compress 750 cu. ft. of air per minute from 14 lb. per sq. in. abs. pressure to 60 lb. per sq. in. abs. pressure in a single-stage compressor; n, 1.25; clearance, 3 per cent; inlet air temperature, 85 deg. F.

6. (*a*) What horsepower would be required to compress the air in problem 5 if two-stage compression were employed with perfect intercooling? (*b*) Calculate the temperature of the air as it entered the intercooler. (*c*) Calculate the heat absorbed (B.t.u. per min.) by the cooling water in the intercooler, assuming no loss by radiation.

7. Find the clearance volume in per cent for a compressor with a volumetric efficiency of 83.5 per cent when air is compressed from 13.2 to 87 lb. per sq. in. abs. pressure along a compression curve of $n = 1.32$.

8. Determine the horsepower required to compress 600 cu. ft. of air per minute from 14 lb. per sq. in. abs. to 100 lb. per sq. in. abs. pressure in a single-stage compressor; n, 1.27; clearance, 4 per cent; and inlet air temperature, 60 deg. F.

9. Find the horsepower required to compress 600 cu. ft. of air per minute from 14 lb. per sq. in. abs. to 100 lb. per sq. in. abs. pressure in a two-stage compressor with perfect intercooling; n, 1.27; clearance, 4 per cent; and inlet air temperature, 60 deg. F.

10. (*a*) Calculate the heat removed (B.t.u. per min.) in the intercooler for the compressor of problem 9. (*b*) Find the theoretical amount (pounds) of water required per minute in the intercooler if the water is available at a temperature of 62 deg. F. and is to be discharged at a temperature of 80 deg. F.

11. Find the cylinder dimensions for the single-stage compressor in problem 8 if it operates at 100 r.p.m. and is double-acting. Piston speed, 450 ft. per min.

12. Find the cylinder dimensions for the two-stage compressor in problem 9 if it operates at 100 r.p.m. and is double-acting. Piston speed, 450 ft. per min.; assume 3-lb. per sq. in. pressure drop in the intercooler.

13. The initial pressure of an air compressor is 13.5 lb. per sq. in. abs. The clearance is 3 per cent and n is 1.22. (*a*) Find the volumetric efficiency if the discharge pressure were 67.5 lb. per sq. in. abs. (*b*) Find the volumetric efficiency if the discharge pressure were 135 lb. abs.

14. An air compressor with a cylinder 10 in. by 12 in. is double-acting and driven at a speed of 150 r.p.m. Clearance is $2\frac{1}{2}$ per cent, n is 1.33, p_1 is 13 lb. per sq. in. abs., and p_2 is 78 lb. per sq. in. abs. (*a*) Find the capacity of the compressor in cubic feet per min. (*b*) Find the i.hp. of the steam engine driving the compressor if the mechanical efficiency is 92 per cent.

15. Determine the relative work for single-stage, two-stage, and three-stage compression between 14 and 210 lb. per sq. in. abs. pressure for a given amount of air entering the condenser. Assume n as 1.25.

16. Find the size of a double-acting air engine to develop 25 hp. when operating at 200 r.p.m. Clearance, 4 per cent; temperature of inlet air, 215 deg. F.; supply pressure, 100 lb. per sq. in. abs.; cut-off at 30 per cent of the stroke; and exhaust pressure, 14.5 lb. per sq. in. abs. Piston speed 400 ft. per min. and $PV^{1.28} = C$.

17. (*a*) What is the temperature of the air at the end of expansion (release pressure) in the engine of problem 16? (*b*) If the air were supplied at 500 deg. F., what would be the temperature at the end of expansion (release pressure)?

18. Air flowing through a Venturi tube had an inlet temperature of 140 deg. F., an inlet pressure of 6 in. of mercury gage, and a throat pressure of $3\frac{1}{2}$ in. of mercury gage. Barometer reading, 29.45 in. of mercury. Inlet diameter of tube, 3 in.; throat diameter, $1\frac{1}{4}$ in. (*a*) How many pounds of air were flowing per minute? Assume temperature of mercury at room temperature of 70 deg. F. (*b*) Find the number of cubic feet of air per minute at suction conditions to the compressor (29.45 in. of mercury and 70 deg. F.).

19. Air was forced through a 1.903-in. diameter orifice placed between the flanges of a 3-in. diameter pipe. The differential pressure across the orifice was 4.2 in. of water. (*a*) Calculate the quantity of air flowing in cubic feet per second. Barometer, 29.15 in. of mercury; temperature of air in the pipe, 90 deg. F. (*b*) Find the velocity of the air through the orifice in feet per second. (*c*) Find the velocity of the air in the pipe in feet per second. (*d*) Where was the downstream pressure connection located? Use curves in Fig. 268.

20. The discharge from an air compressor was at a pressure of 60 lb. per sq. in. gage and flowed through a 3-in. diameter Compressed Air Society nozzle into the atmosphere. The observed manometer reading was 5 in. of water; temperature of the liquid in the manometer, 70 deg. F.; temperature of air entering the nozzle, 90 deg. F.; barometer, 28.38 in. of mercury. How many cubic feet of air were discharged per minute, if the nozzle coefficient was 0.98? Test apparatus was arranged as shown by Fig. 269.

CHAPTER XIX

INTERNAL-COMBUSTION ENGINES

291. Principles of Operation.—Some of the heat generated by the combustion of fuel in a furnace passes through the metal plates or tubes of the boiler to the water, and the steam generated may be used in an engine cylinder or turbine to do work. The working medium is therefore entirely different from the products of combustion of the fuel. This intermediate step, with its numerous heat losses, is not necessary in an internal-combustion engine, as combustion of the fuel takes place within the cylinder and work is done by expansion of the resulting gases. The working medium, therefore, undergoes a chemical change during this operation, and the thermodynamic process is not cyclic. In the early stages, before combustion, the medium is a mixture of fuel with air; generally an excess of air is required to provide enough oxygen for complete combustion. In the later stages, after combustion, there is a mixture of the products of combustion with nitrogen and any surplus of air. The fuel commonly enters as a gas or vapor and is drawn in with a suitable proportion of air; but the fuel may be injected into the cylinder as a liquid, becoming vaporized after admission or being burned as it enters.

Typical examples of internal-combustion engines are the ordinary gas and gasoline engines, in which a " charge " of air mixed with combustible gas or vaporized liquid fuel is drawn into the cylinder by the piston. The mixture is compressed into a clearance space and there ignited by an electric spark, so that explosive combustion takes place while the volume of the charge is nearly constant. The heat thus internally developed gives the working medium a high temperature and pressure; expansion of the gases occurs, and work is done as the piston advances.

Considered thermodynamically, internal-combustion engines have the advantage, over the steam engine and turbine, that the working medium " takes in " heat (by its own combustion) at a very high temperature. In the combustion of the charge a temperature of 4000 deg. F. or more is reached. The full thermodynamic advantages of a high initial temperature cannot be realized in practice, for if the cylinder walls were allowed to reach this high temperature they would soon be destroyed. Lubrication of the piston would also be impossible, to say

nothing of the probable premature ignition of the incoming fuel mixture; hence the cylinder is generally water-jacketed to keep the walls and parts cool. With large gas and oil engines the difficulty is to keep the cylinder, piston, and valves cool; with steam engines, the cylinder should be kept hot to reduce the losses due to condensation of the steam. The average temperature at which the heat is developed in an internal-combustion engine is far above that at which heat is received by the working medium of a steam engine or turbine. On the other hand, internal-combustion engines do not discharge heat at so low a temperature as do steam engines and turbines. But the actual working range of temperature is so large that a gas engine can convert into work a larger fraction of the heat energy of the fuel than is utilized by any steam engine or steam turbine. A good internal-combustion engine will convert about 35 to 40 per cent of the energy of the fuel into work; the best steam engines convert no more than about 20 per cent, and the best steam turbines convert no more than about 30 per cent of the energy of the fuel into work.

The small internal-combustion engine has made possible the automobile, tractor, and airplane. In plants having combustible by-product gases, such as those from blast furnaces and coke ovens, large-size units are frequently used. Internal-combustion engines are well adapted for use in marine service and small plants, where space is of prime importance.

292. Classification.—Internal-combustion engines may be classified according to: *cycles of combustion*—Otto cycle, combustion at constant volume, and Diesel cycle, combustion at constant pressure; *cycles of operation*—two-stroke cycle and four-stroke cycle; *number of cylinders*—one, two, three, four, etc., or single-cylinder and multicylinder; *arrangement of cylinders*—vertical or horizontal, single-acting or double-acting; *uses*—stationary, portable, marine, automobile, tractor, airplane, etc.; *fuels employed* and how the fuel is supplied to the engine cylinder, as gas, oil, gasoline, kerosene, etc., and carburetor, hot-bulb, solid-injection, air-injection, etc.

No single system of classification serves for all engines; on the other hand, it is not necessary to give all of the above characteristics to define an engine. If an engine is said to be an automobile engine, it is partially classified, and would be completely so, if information regarding the arrangement and number of cylinders were given. A mental picture is immediately formed of a four-cycle Otto engine using gasoline as fuel. However, a complete description would be as follows: Otto cycle, six-cylinder, single-acting, vertical, four-stroke cycle, gasoline, carburetor, 60-brake hp. at 2500 r.p.m. A certain type of engine is called a marine engine, but the engine can be more completely identified

thus: Diesel, four-cylinder, double-acting, vertical, two-stroke cycle, heavy fuel-oil, solid-injection, 300 brake hp. at 350 r.p.m.

293. Cycles of Combustion.—There are two distinct methods by which the combustion of a fuel may be accomplished in an internal-combustion engine.

(*a*) Air, intimately mixed with a suitable quantity of fuel in gaseous or vaporized form, is compressed to a moderate extent in the cylinder, and is then ignited by an electric spark, the resultant rise of pressure being practically instantaneous, with nearly constant volume.

(*b*) Air only is compressed in the cylinder, the compression being carried to a high pressure so that the resultant temperature is sufficient to ignite the fuel. At the instant this high compression pressure is attained, the fuel is sprayed into the cylinder; the piston at the same time begins to move downward on the power stroke. The ignition of the fuel tends to produce a further increase of temperature and so maintains a practically constant pressure during the first portion of the downward stroke of the piston. The period of fuel injection can be prolonged and the pressure maintained until the available oxygen is exhausted.

The essential difference between *a* and *b* will be at once apparent. Engines of class *a*, which include all gas and gasoline engines besides many oil engines, get their power impulse from a sudden rise of pressure, so rapid as to be called *explosion*, *constant-volume* or *Otto-cycle* engines. The operation of engines of class *b* depends upon the maintenance of constant pressure during the period of fuel injection and combustion. This type is practically confined to heavy fuel oils; the engines are commonly known as *constant-pressure* or *Diesel engines.*

294. Cycles of Operation. The process of introducing the working medium into the cylinder, compressing, igniting, and driving it out of the cylinder after having done work, will now be considered. This cycle of events may be accomplished in one of two ways: by the four-stroke cycle or by the two-stroke cycle; other cycles have been attempted, but none have met with any measure of success.

Four-stroke cycle.—In the most usual type of internal-combustion engine, the cycle of operation (or mechanical cycle) is completed in four strokes of the piston or two revolutions of the crankshaft. (1) The charge of gas and air may be drawn into the cylinder, *A* in Fig. 271, the intake valve being open on the downward (suction) stroke of the piston. (2) At the beginning of the return (compression) stroke, *B* in Fig. 271, intake and exhaust valves being closed, the mixture of air and gas is compressed into the clearance space varying generally from one-fourth to one-eighth of the original volume. Near the head-end dead center the compressed mixture is ignited by an electric spark, heating the charge

to a very high temperature and consequently raising the pressure. (3) During the next downward stroke the fired mixture expands, *C* in Fig. 271, doing work on the piston which, through the connecting rod and crank, creates a turning effort of the crankshaft, as indicated. (4) As the piston moves upward on the fourth stroke, the products of combustion are forced out of the cylinder, *D* in Fig. 271, the exhaust valve being open. A small quantity of the burnt mixture remains in the clearance space and is mixed with the next charge of fuel, unless special means are used to remove these inert and non-combustible

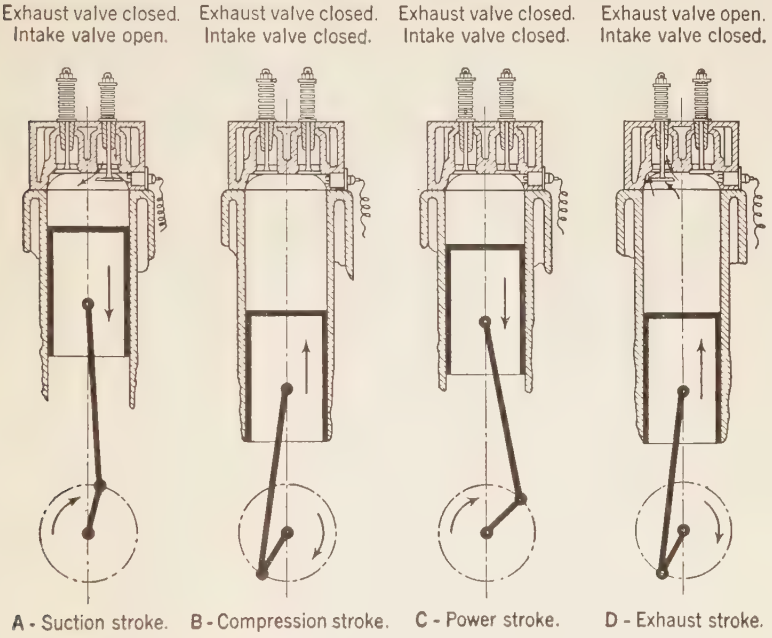


FIG. 271.—Cycle of operation in a four-stroke Otto cycle engine.

gases by *scavenging*.[1] After this cycle the inlet valve is again opened, and the cycle of operations is repeated. In the four-stroke cycle there is only one power stroke out of every four, which seems retrogressive when compared with steam engine practice; nevertheless, the four-stroke cycle has entirely justified itself.

The indicator diagram of the four-stroke cycle is shown by Fig. 272. During the suction stroke *0 1*, the pressure is below atmospheric; and during exhaust *4–0*, the pressure is above atmospheric. The area of the

[1] Scavenging may be defined as any action in which the products of combustion are forced out of the cylinder of an internal-combustion engine by the admission of fresh air.

inclosure *0–1–5* represents the work done *on* the gases during the cycle and is due to the " pumping action " of the piston in forcing out the exhaust gases (high pressure at *4*) and drawing in the charge during suction (low pressure along *0 1*). The curve *1–5–2* represents compression of the charge, V_c being the volume of the clearance space into which the charge is compressed. Line *2–3* is the rise of pressure caused by the explosion, and *3–4* is the expansion which constitutes the effective working stroke. The area *2–3–4–5* represents the work done *by* the gases during the cycle, and the net work developed during the four-stroke cycle is, therefore, (*2–3–4–5*) minus (*0–1–5*).

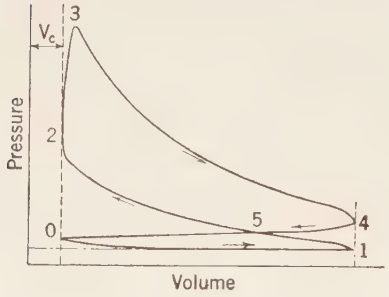


Fig. 272.—Indicator diagram of the four-stroke Otto cycle.

It is to be noted that the four-stroke principle can be used for engines operating on the Diesel cycle, as well as for Otto engines (as described above).

Two-stroke cycle.—Figure 273 illustrates the action of an internal-combustion engine in which the completion of one cycle of operation is

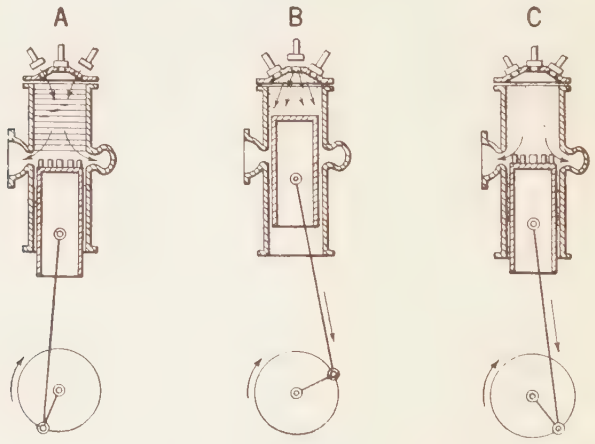


Fig. 273.—Cycle of operation in a two-stroke Diesel cycle engine.

accomplished in two strokes of the piston. At *A* the scavenging valves in the cylinder head are open, and a charge of fresh air (compressed by an air compressor) blows into the cylinder, forcing out the exhaust gases as indicated. The scavenging valves close, and as the piston moves upward the exhaust ports are covered. Continued upward motion

compresses the air charge until upper dead center is reached. The clearance volume is small, and the maximum or final compression pressure rises to about 500 lb. per sq. in. The work done on the air charge during compression causes the temperature to rise to 1000 deg. F. or more. Just before the piston reaches top dead center a charge of fuel is injected into the cylinder and because of the high temperature combustion occurs as the piston starts downward, as at *B* in Fig. 273. The piston continues to the end of its stroke under the influence of the expanding gases. Before the completion of the downward stroke, as at *C*, the piston uncovers passages or ports in the side of the cylinder through which the exhaust gases pass. By this means one explosion is

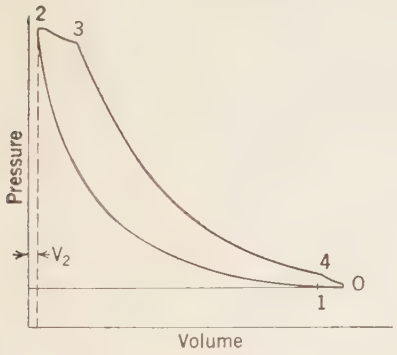


Fig. 274.—Indicator diagram of the two-stroke Diesel engine.

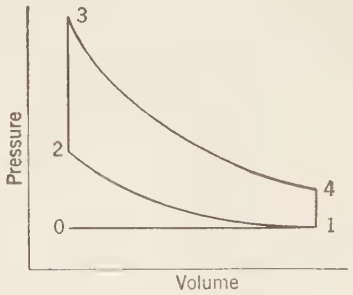


Fig. 275.—Indicator diagram of the ideal Otto cycle.

obtained every revolution of the engine shaft when using one single-acting cylinder.

An indicator diagram of a two-stroke cycle engine is shown by Fig. 274. A charge of fresh air is admitted under pressure along *0–1*; compression of this air occurs along *1–2* with a resultant high pressure and temperature. Injection of fuel and combustion at nearly constant pressure are represented by line *2 3*. Expansion of the gases takes place along *3 4*. From *4* to *0*, exhaust occurs and scavenging of the burnt gases in the cylinder is effected.

Although the two-stroke cycle was described above for an engine operating on the Diesel principle, the two-stroke cycle is also applicable to Otto engines.

295. The Otto Cycle.—The indicator diagram of the ideal Otto cycle is shown by Fig. 275. The line *0 1* (atmospheric pressure) indicates the entrance of the charge of fuel and air during the first stroke; the curve *1–2*, adiabatic compression of this charge on the return stroke. The line *2–3* represents the rise of pressure at constant volume when the

charge is ignited and the fuel burned; and the curve *3–4*, adiabatic expansion of the products of combustion. At point *4* (end of the expansion stroke) the exhaust valve opens with the result that part of the gas escapes at once and the pressure falls to that of the atmosphere, giving line *4–1*. Line *1–0* represents the exhaust stroke by which the cylinder is emptied of burned gases, preparatory to receiving a fresh charge during the first stroke of the next cycle. If the ideal engine were of the two-stroke cycle type, line *0–1* would be omitted from the indicator diagram.

For the purpose of obtaining a simple standard with which actual engines may be compared, a practice has been adopted of treating the working medium as air, to which, between *2* and *3*, Fig. 275, there is imparted a definite quantity of heat, namely, the heat of combustion of the charge. This standard, known as the "Air Standard," is a measure of the ideal thermal efficiency of an internal-combustion engine. Such an ideal efficiency affords a useful standard for comparison with the performance of an actual engine. To simplify the anlysis, certain conditions are assumed for this ideal air standard Otto cycle: that the mixture of gas and air is admitted during the suction stroke at atmospheric pressure; that compression is adiabatic, which necessitates the assumption that the cylinder walls and piston are non-conductors of heat; that combustion is instantaneous and complete, and takes place at constant volume; that expansion is adiabatic; that the products of combustion are exhausted at atmospheric pressure, and no exhaust gases left in the clearance space; that the specific heat is constant throughout the cycle (independent of temperature as well as pressure); that, in the case of a liquid fuel, the fuel is completely vaporized before entering the cylinder.

The lines *1–2* and *3–4*, Fig. 275, are assumed to be adiabatics. All the heat must therefore be supplied along line *2–3* and rejected along *4–1*. The *heat supplied* along *2–3* is

$$_2H_3 = Mc_v(T_3 - T_2), \tag{167}$$

in which M = weight of the charge, lb. per cycle.
c_v = specific heat at constant volume, 0.173 for air.
T_3 = absolute temperature at *3*, deg. F.
T_2 = absolute temperature at *2*, deg. F.

The *heat rejected* along *4–1* is

$$_4H_1 = Mc_v(T_4 - T_1), \tag{168}$$

in which T_4 = absolute temperature at *4*, deg. F.
T_1 = absolute temperature at *1*, deg. F.

The work done is

$$_2H_3 - {}_4H_1 = Mc_v(T_3 - T_2) - Mc_v(T_4 - T_1).$$

The air standard thermal efficiency of the Otto cycle is:

$$\frac{{}_2H_3 - {}_4H_1}{{}_2H_3} = \frac{Mc_v(T_3 - T_2) - Mc_v(T_4 - T_1)}{Mc_v(T_3 - T_2)},$$

$$= \frac{(T_3 - T_2) - (T_4 - T_1)}{(T_3 - T_2)},$$

$$e_a = 1 - \frac{T_4 - T_1}{T_3 - T_2}. \tag{169}$$

Then, by Art. 16

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} \quad \text{and} \quad \frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1}.$$

But

$$V_3 = V_2 \quad \text{and} \quad V_4 = V_1, \text{ as shown in Fig. 275.}$$

Hence

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} \quad \text{and} \quad \frac{T_4}{T_3} = \left(\frac{V_2}{V_1}\right)^{k-1};$$

and

$$\frac{T_1}{T_2} = \frac{T_4}{T_3} \quad \text{or} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}.$$

Then

$$\frac{T_4}{T_1} - 1 = \frac{T_3}{T_2} - 1 \quad \text{and} \quad \frac{T_4 - T_1}{T_1} = \frac{T_3 - T_2}{T_2},$$

or

$$\frac{T_4 - T_1}{T_3 - T_2} = \frac{T_1}{T_2}. \tag{170}$$

Substituting equation 170 in equation 169,

$$e_a = 1 - \frac{T_1}{T_2}. \tag{171}$$

But

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}, \quad \text{also} \quad \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} \quad \text{(see Art. 16).}$$

Therefore

$$e_a = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}; \tag{172}$$

and

$$e_a = 1 - \left(\frac{V_2}{V_1}\right)^{k-1}. \tag{173}$$

Equation 171 shows that the air standard thermal efficiency of a gas engine employing the Otto cycle depends upon the ratio of the absolute temperatures before and after compression. From equation 173, the smaller the clearance, V_2, of the engine, the larger is the efficiency. Equation 172 is important as showing the beneficial influence of compression, the efficiency increasing as P_2 increases, since the pressure P_1 remains nearly constant.

Example.—In an engine operating on the ideal Otto cycle the suction pressure is 14.7 lb. per sq. in. abs.; temperature at the beginning of compression 140 deg. F.; highest pressure attainable (at the end of combustion), 350 lb. per sq. in. abs. at a temperature of 2140 deg. F. Find: (*a*) the pressure at the end of compression, (*b*) the temperature at the end of compression, (*c*) the air standard thermal efficiency, and (*d*) the clearance volume in per cent of the stroke. Assume $k = 1.40$.

Solution.—

(*a*) From equation 36,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}},$$

$$T_2 = 600\left(\frac{P_2}{14.7}\right)^{\frac{0.4}{1.4}}. \tag{A}$$

From equation 16,

$$\frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} \quad \text{and} \quad V_2 = V_3; \text{ see Fig. 275.}$$

Then

$$T_2 = 2600\left(\frac{P_2}{350}\right). \tag{B}$$

Putting T_2 of (A) in place of T_2 in (B),

$$600\left(\frac{P_2}{14.7}\right)^{\frac{2}{7}} = 2600\left(\frac{P_2}{350}\right).$$

Hence

$$P_2 = (37.44)^{1.4} = 160 \text{ lb. per sq. in. abs.}$$

(*b*) $T_2 = 2600(\frac{160}{350}) = 1188$ deg. abs. or 728 deg. F.

(*c*) $e_a = 1 - \frac{T_1}{T_2} = 1 - \frac{600}{1188} = 1 - 0.505 = 0.495$ or 49.5 per cent.

(*d*) From equation 35,

$$P_1V_1^k = P_2V_2^k;$$

therefore,

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k.$$

Take the piston displacement = 1.00, then

$$V_1 = V_2 + 1.$$

Thus

$$\frac{160}{14.7} = \left(\frac{V_2 + 1}{V_2}\right)^k,$$

and $V_2 = 0.222$ or 22.2 per cent of the stroke.

With increased pressure ratios $\left(\frac{P_2}{P_1}\right)$, the air standard efficiency of the ideal Otto cycle increases as shown in Table XX, with $k = 1.4$.

TABLE XX

Effect of Compression on Efficiency of Ideal Otto Cycle

P_1, Lb. per Sq. In. Abs.	P_2, Lb. per Sq. In. Abs.	$\frac{P_2}{P_1}$	Air Standard Thermal Efficiency	Clearance Required
14.7	29.4	2	0.179	
14.7	58.8	4	0.327	0.591
14.7	88.2	6	0.401	0.386
14.7	117.6	8	0.448	0.293
14.7	147.0	10	0.483	0.239
14.7	176.4	12	0.508	0.204
14.7	205.8	14	0.530	0.179
14.7	235.2	16	0.547	0.160
14.7	294.0	20	0.575	0.133
14.7	441.0	30	0.622	0.097

Table XX shows that there is at first a very rapid gain of efficiency with increased compression, but when the compression is high the thermodynamic advantage of increasing the pressure is small. The efficiencies actually obtained in tests of engines are considerably lower, usually 40 to 60 per cent less, owing to heat losses, variation in specific heat, and to the fact that combustion is not instantaneous.

With all engines of this class there is a practical limit to the amount of compression. The pressure must not be so high as to cause pre-ignition by unduly raising the temperature of the mixture during the compression stroke, nor so great as to give to the explosion, when it does occur, the peculiarly sudden character known as detonation. *Detonation* is the nearly simultaneous explosion of all parts of an explo-

sive mixture by means of a compression wave, as distinguished from an explosion in which ignition spreads from one part to another in a manner like that of slow combustion. It occurs through the development of a compression wave whose pressure heats the still unburnt portions of the charge to a temperature at which they suddenly combine, without waiting to be reached by the igniting flame. Detonation or " pinking " may be distinguished from normal explosion by its sharp, metallic, hammer-like, ringing sound, produced when the high-pressure compression wave strikes and rebounds from the cylinder walls. When a quarter per cent by volume of tetraethyl lead, $Pb(C_2H_5)_4$, is added to ordinary gasoline, the compression pressure may safely be increased when there is active turbulence, and when the combustion chamber is so designed as to avoid any long distance in the travel of ignition from the spark plug.

The compression pressures for engines operating on the Otto cycle vary from 40 to 200 lb. per sq. in., differing with various kinds of fuels, for example, being low when there is much hydrogen and comparatively high for alcohol. For gasoline automobile engines, the pressures range from 60 to 100 lb. per sq. in. Difficulty may be experienced with automobile engines designed for the higher compression pressures when operated with ordinary gasoline. With producer-gas as a fuel, compression pressures of 100 to 150 lb. per sq. in. are allowable; and with blast-furnace gas, pressures as high as 180 lb. per sq. in. may be used. When air alone is compressed, as in the Diesel engine, and the fuel is injected only when combustion is intended to occur, the same considerations do not apply; compression may then be usefully carried much higher.

Comparison of Fig. 272 and Fig. 275 shows the variation of the actual Otto cycle from the theoretical. In the actual engine, combustion is not instantaneous but begins before the piston reaches the end of the compression stroke, at 2, in Fig. 272. The exhaust valve is generally opened before the end of the expansion stroke, in order that the pressure in the cylinder may be practically atmospheric before the piston starts on the exhaust stroke. The exhaust stroke, Fig. 272, is at a pressure slightly above atmospheric, and during the suction stroke the pressure in the cylinder is below atmospheric. The area of this loop, representing a loss of work, is usually small, especially if ample valve areas are provided and the engine is operated at a relatively low speed. The compression and expansion curves of the actual card are not true adiabatics, principally because of the heat loss to the water jacket and variable specific heat; the curves, however, approximate very closely the curve $PV^n = C$. For a definite value of n, the discussion and equations given in this article apply for any gas. The value of n for gasoline

mixtures is generally taken as 1.32 for both the compression and expansion curves.

296. The Diesel Cycle.—The ordinary gas engines, automobile engines, and many oil engines are of the constant-volume (Otto) type of internal-combustion engine. If the pressure of the working medium does not change while combustion is taking place, the cycle is called the constant-pressure or Diesel cycle. Suppose that the air is separately compressed into the clearance space before any fuel is admitted and that the fuel (oil) is then forced into the cylinder, burning as it enters, while the piston begins its outward (expansion) stroke. By suitably regulating the rate of admission of the fuel, the pressure may be kept constant until combustion is completed, *2–3*, Fig. 276. By avoiding any admixture of fuel with the air in the cylinder, before and during compression, a high compression pressure is practicable with the Diesel cycle with no possibility of pre-ignition, and consequently a high thermal efficiency is generally obtained.

In the ideal Diesel-engine cycle, Fig. 276, air enters the cylinder along *0–1*, the air is compressed on the return stroke of the piston along the adiabatic curve *1–2*. The air is compressed to about 500 lb. per sq. in. or more before the fuel is admitted. At *2* the fuel is forced into the cylinder, and, on account of the high temperature of the compressed air, the fuel ignites. The supply of fuel continues as the piston moves back, causing combustion at constant pressure. The curve *3–4* represents the adiabatic expansion of the products of combustion. At point *4* the exhaust valve opens and the pressure drops to *1* (atmospheric). On the fourth stroke, along *1–0*, the products of combustion are forced through the exhaust opening; some of the products, however, remain in the clearance space and are mixed with the incoming air on the first stroke of the next cycle.

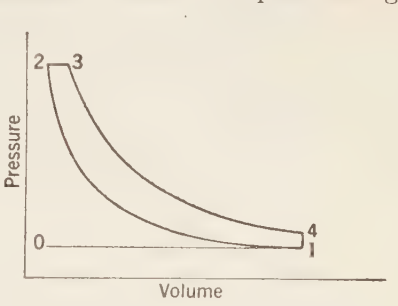


Fig. 276.—Indicator diagram of the ideal Diesel cycle.

To simplify the analysis, certain conditions are assumed for the ideal air standard cycle of the Diesel engine: that the pressure of the air during the suction stroke is atmospheric; that compression is adiabatic, which necessitates the assumption that the cylinder walls and piston are non-conductors of heat; that the combustion of the fuel is at constant pressure, that is, the line *2–3*, Fig. 276, is horizontal; that, in the case of liquid fuel, the fuel is completely vaporized before entering the cylinder; that the fuel is injected into the cylinder without admixture

of air, and that it enters the cylinder at the temperature T_1; that expansion is adiabatic; that the products of combustion are exhausted at atmospheric pressure, and no exhaust gases are left in the clearance space; that the specific heat is constant throughout the cycle (independent of temperature as well as pressure).

Since the lines *1–2* and *3–4*, Fig. 276, are assumed adiabatics, all the heat of the cycle must therefore be supplied along line *2–3* and rejected along *4–1*. The *heat supplied* along *2–3* is

$$_2H_3 = Mc_p(T_3 - T_2), \tag{174}$$

in which M = weight of the air in the cylinder, assumed as constant since the fuel is oil and the increase in weight of the charge is small, lb. per cycle.
c_p = specific heat at constant pressure, 0.242 for air.
T_3 = absolute temperature at *3*, deg. F.
T_2 = absolute temperature at *2*, deg. F.

The *heat rejected* along *4–1* is

$$_4H_1 = Mc_v(T_4 - T_1), \tag{175}$$

in which c_v = specific heat at constant volume, 0.173 for air.
T_4 = absolute temperature at *4*, deg. F.
T_1 = absolute temperature at *1*, deg. F.

The work done is

$$_2H_3 - {}_4H_1 = Mc_p(T_3 - T_2) - Mc_v(T_4 - T_1).$$

The air standard thermal efficiency of the Diesel cycle is:

$$\frac{{}_2H_3 - {}_4H_1}{{}_2H_3} = \frac{Mc_p(T_3 - T_2) - Mc_v(T_4 - T_1)}{Mc_p(T_3 - T_2)},$$

$$e'_a = 1 - \frac{(T_4 - T_1)}{k(T_3 - T_2)}. \tag{176}$$

Since *2–3* is at constant pressure and *4–1* at constant volume

$$T_3 = T_2\left(\frac{V_3}{V_2}\right), \tag{177}$$

and

$$T_4 = T_1\left(\frac{P_4}{P_1}\right). \tag{178}$$

As *1–2* and *3–4* are adiabatics

$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^k \quad \text{and} \quad \frac{P_4}{P_3} = \left(\frac{V_3}{V_4}\right)^k;$$

by dividing, and since $V_1 = V_4$,

$$\frac{P_4}{P_1} = \left(\frac{V_3}{V_2}\right)^k. \tag{179}$$

Substituting (179) in (178),

$$T_4 = T_1\left(\frac{V_3}{V_2}\right)^k. \tag{180}$$

Substituting (180) and (177) in (176),

$$e'_a = 1 - \frac{T_1}{T_2}\left\{\frac{\left[\left(\frac{V_3}{V_2}\right)^k - 1\right]}{k\left(\frac{V_3}{V_2} - 1\right)}\right\}. \tag{181}$$

Since T_1 and T_2 are unknown in most cases a ratio of volumes and pressures will usually be more convenient for finding the air standard efficiency of the Diesel-engine cycle. Thus

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}.$$

Hence

$$e'_a = 1 - \left(\frac{V_2}{V_1}\right)^{k-1}\left\{\frac{\left[\left(\frac{V_3}{V_2}\right)^k - 1\right]}{k\left(\frac{V_3}{V_2} - 1\right)}\right\}; \tag{182}$$

and

$$e'_a = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}\left\{\frac{\left[\left(\frac{V_3}{V_2}\right)^k - 1\right]}{k\left(\frac{V_3}{V_2} - 1\right)}\right\}. \tag{183}$$

Equation 183 shows that the thermal efficiency of the Diesel cycle depends not only upon the compression, but also upon the ratio of the volume (V_3) at cut-off to the clearance volume (V_2). The efficiency increases as this ratio decreases.

Example.—In an engine operating on the ideal Diesel cycle the suction pressure is 14.5 lb. per sq. in. abs., and the fuel is injected for 4 per cent of the stroke.

The clearance volume is 6 per cent of the stroke. Find the air standard thermal efficiency, assuming $k = 1.4$.

Solution.—From equation 182,

$$e'_a = 1 - \left(\frac{0.06}{1.06}\right)^{0.4} \left\{ \frac{\left[\left(\frac{0.10}{0.06}\right)^{1.4} - 1\right]}{1.4\left(\frac{0.10}{0.06} - 1\right)} \right\};$$

$$e'_a = 1 - \frac{1.05}{3.15 \times 0.933} = 0.642 \quad \text{or} \quad 64.2 \text{ per cent.}$$

Comparison of Figs. 274 and 276 shows the deviation of the actual diagram from the theoretical. In the actual engine, compression approaches the adiabatic curve since practically pure air is compressed in the cylinder, and the value of exponent k is usually between 1.35 and 1.40. In general, the fuel cannot be injected rapidly enough to maintain a constant pressure, and line *2–3*, Fig. 274, shows a slight decrease during combustion of the fuel. The value k for the expansion curve of the actual engine is about 1.35 or less.

297. Losses in Internal-combustion Engines.—The total amount of heat supplied during an internal-combustion-engine cycle may be accounted for as follows: Heat converted into work; heat used in overcoming friction; heat carried away in the exhaust gases; heat given to the water jacket; and heat lost by radiation, leakage, etc.

There are wide variations in the relative proportions of the above losses, depending upon the type, size, and operating conditions of the engine under consideration. For an automobile engine operating on the Otto cycle, the distribution of heat may be: heat converted into work about 20 per cent, lost to friction 8 per cent, carried away in the exhaust gases 34 per cent, given to the water jacket 28 per cent, and radiation and other losses 10 per cent. For Diesel engines, the distribution of heat may be: converted into work about 30 per cent, lost to friction 12 per cent, carried away in exhaust gases 28 per cent, given to water jacket 25 per cent, and radiation and other losses 5 per cent.

The heat carried away in the exhaust gases is generally the largest loss, resulting from incomplete expansion and exhaust at a high temperature (600 to 1200 deg. F.). This unavoidable loss can be reduced by carrying the expansion down to a lower pressure, but mechanical difficulties are encountered. The high temperature attained during explosion results in a very rapid extraction of heat by the cylinder walls. The higher the temperature the more rapid is this loss through the cylinder walls to the water jacket. Cooling is necessary as a protection for the cylinder walls, to facilitate lubrication, and to prevent

pre-ignition. Because of the higher pressure employed in Diesel engines the moving parts are heavier than in Otto cycle engines of equal capacity, and hence the frictional losses are higher in Diesel engines.

298. Internal-combustion-engine Fuels.—The principal products of nature convertible into use as fuel for internal-combustion engines are coal, natural gas, and petroleum oil. Some of the artificial or manufactured fuels available are illuminating gas, blast-furnace gas, coke-oven gas, producer gas, alcohol, petroleum distillates, residual liquid fuels, and blends of distillate and residual fuels.

Internal-combustion engines may be operated on a gaseous fuel, as well as any other fuel which can be transformed into a vapor or gas. Gasoline, kerosene, fuel oil, or alcohol may be used in the engine cylinder without the intermediate step of converting the fuel into a gas.

Natural gas.—This fuel is taken directly from gas wells, and usually issues with the oil from oil wells. The chemical constituents vary widely in different localities, and also even in the gas taken from the same region; it is composed mainly of methane (CH_4), and in some cases there is considerable hydrogen. Any gas containing a large amount of methane or hydrogen will withstand only low compression without pre-ignition; some of the leaner gases, such as blast-furnace and producer gas, will allow higher compression pressures. The high heat value of natural gas ranges from 800 to 1150 B.t.u. per cu. ft. It has been conveyed long distances through welded pipelines, because of its convenience, excellent properties, and low cost.

Illuminating or water gas.—This gas is usually made by passing steam over white-hot coke. When the gas is made from coke its composition is less liable to variation than when coal, with its varying hydrogen content, is employed. The gas is clean, and the high heat value averages 450 to 600 B.t.u. per cu. ft. As commonly produced this gas is too expensive (seldom less than 75 cents per 1000 cu. ft.) for extensive use in a gas engine; it is used only for small engines or for engines that operate infrequently.

Blast-furnace gas, or "downcomer" gas, is a by-product of combustion in the operation of a blast furnace. It may be used to preheat the air blast and for various other purposes in the reduction of iron ore. At some blast-furnace plants, the gas from the furnace is used for fuel in the gas engines that drive the blowers, the latter supplying the air necessary to support combustion in the blast furnace. The principal combustible in blast-furnace gas is carbon monoxide (CO), which averages 30 per cent; the high heat value ranges from 80 to 120 B.t.u. per cu. ft.

Coke-oven gas is a by-product from the manufacture of industrial coke. About one-half of the distilled gas is used for heating the coke

ovens; that excess gas is led away and cooled, then washed. This gas makes an excellent engine fuel and is used as such in localities where it is available. Coke-oven gas contains about 50 per cent hydrogen (H_2) and 35 per cent methane (CH_4), and has a high heat value of 400 to 600 B.t.u. per cu. ft.

Producer gas.—This gas is formed by the combustion of any grade of coal, coke, or peat with a deficient air supply. Incomplete combustion results, with the formation of a considerable quantity of carbon monoxide (CO). Producer gas has a high heat value of 120 to 160 B.t.u. per cu. ft.

Liquid fuels may be divided into five general classes: crude petroleum or crude oil, gasoline, kerosene, fuel oil, and alcohol. *Crude petroleum*, because it is a mixture of chemical compounds of very divergent physical properties, is not a satisfactory fuel; those engine conditions which are favorable for burning one part of the oil are not necessarily favorable for the other constituents. Further, crude oil is not used to any great extent as it contains valuable light fuels which can be distilled off, leaving behind a fuel-oil residue. The high heat value of crude oil, gasoline, kerosene, and fuel oil is generally from 19,000 to 20,500 B.t.u. per lb.

The greatest demand is for *gasoline*; and the inability to supply that demand by the distillation process, the product of which is termed "straight-refinery gasoline," has led to the extensive development of cracking processes and the impregnating of the heavier liquids with the ethers to give fuels with more or less suitable properties for use in automobile and similar engines. Under high-temperature and high-pressure conditions, the heavier hydrogen-carbon compounds break down into lighter and more volatile hydrocarbons, with deposition of free carbon and in some cases the liberation of a gas. This process is known as cracking and is used in the production of a considerable portion of the gasoline sold in the United States. About 60 per cent of the original volume of crude oil can now be blended into gasoline as a result of improved refining processes.

Kerosene is also distilled from petroleum, but it is heavier and less volatile than gasoline and safer to use and to transport. Some of the disadvantages are: cold engine will not start readily with this fuel; kerosene is not easily vaporized at normal temperatures; and it has a tendency to detonate.

Gasoline is that portion of the crude oil distilled off from 100 to 400 deg. F.; kerosene is given off between 400 and 550 deg. F. After the kerosene has been driven off, the dense petroleum which remains is called *fuel oil*. This oil is cheap, safe to handle because of low volatility, and especially adapted for use in Diesel engines.

Alcohol, C_2H_6O, may be manufactured from such by-products as "black-strap" (cane) molasses, waste liquor from sulphite paper-pulp manufacture, sawdust, straw, cornstalks, etc. The high price of gasoline in foreign countries in which there is abundant raw material from which to produce alcohol, but no developed petroleum reserve, has led to an increased use of alcohol or alcohol-gasoline mixtures as an engine fuel. Alcohol is only 60 per cent as rich in heat units as gasoline and is 15 to 20 per cent heavier. Commercial alcohol contains approximately 10 per cent of water by weight and has a high heat value of 11,000 to 13,000 B.t.u. per lb. The range of distillation temperatures is from 160 to 175 deg. F. Alcohol requires special carburetor construction to permit starting an engine cold; but it has highly desirable anti-knock properties, when operating at high compression pressures, with a resultant increase in thermal efficiency.

Probable high prices for petroleum in the future, together with a knowledge of the limitations of oil reserves, have served to create an interest in the possible commercial development of the vast oil-shale resources. The U. S. Geological Survey and others have studied the shale deposits in Colorado, Wyoming, and Utah. Estimates indicate that the shales in this region are capable of producing the surprising total of one hundred billion barrels of crude shale oil. No large-scale production of shale oil is under way, but considerable experimentation in the laboratory, and in full-sized retorts, has been done.

299. The Gasoline Carburetor.—The fuel mixture for Otto-cycle engines is a gas or fuel vapor mixed with air. Hence, with liquid fuels, special devices are required to convert these fuels into gases or vapors. Such devices are indiscriminately known as vaporizers, carburetors, mixers, or mixing valves. The term vaporizer is usually employed when considerable heat for gasification or vaporization is required, as in the case of fuel oil and crude oil. A *carburetor* is a device in which the correct air-fuel mixture is formed, at all loads and speeds of the engine, by passing air over or through the gasoline, kerosene, or fuel oil. When no special heating of the fuel is done, these devices are usually applicable to the more volatile fuels only, as gasoline. The name carburetor, however, may also be used when in this device gasoline is mechanically atomized or sprayed into the current of incoming air. Better results are obtained with complete vaporization of the fuel, but evaporation of the fuel is not necessary if the liquid particles are finely divided and thoroughly mixed with air.

Figure 277 shows line diagrams of a plain tube carburetor. The tip of the nozzle G is placed slightly higher than the normal liquid level in the float chamber in order to avoid leakage of fuel when there is no air flow. When air is flowing through the carburetor the resulting vac-

uum in the manifold causes the fuel to rise to the top of the tube G and discharge into the stream of air. The mixture flows past the throttle valve T (when open) and then through the manifold to the cylinder. The level of gasoline is maintained constant in float chamber F by means of a float which operates the needle valve. As the level falls, the needle valve opens, allowing more gasoline to enter the float chamber. Atmospheric pressure exists on top of the gasoline surface in the float chamber, being produced by means of an air vent. The difference between the atmospheric pressure in the float chamber and the pressure around the top of the tube causes the gasoline to flow into the air stream. A carburetor of this type would give a " leaner " mixture

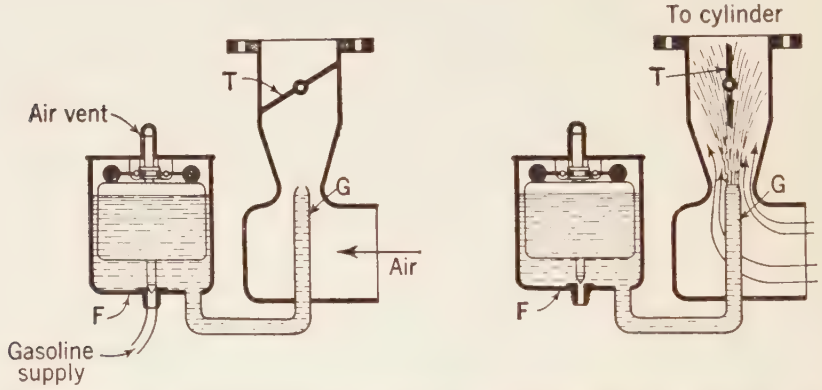


FIG. 277.—Plain tube carburetor; left, engine idle; right, engine in operation with wide-open throttle.

as the engine speed or air flow is increased. Most commercial carburetors are designed to overcome this tendency, and to give the desired air-fuel ratios over the entire range of engine speed.

The Stromberg carburetor, Fig. 278, is a typical plain tube gasoline carburetor provided with special features to insure a lean mixture at normal speeds and to supply the richer mixture necessary for maximum power and high speed. Fuel enters through the gasoline inlet and flows through the gasoline strainer and needle valve seat into the float chamber. The float automatically maintains the correct fuel level in the carburetor. The fuel then flows through the main metering jet to the main discharge-jet or to the idle discharge-jet, depending on the position of the throttle valve. At speeds less than 12 miles per hour the throttle valve is practically closed and the gasoline enters through the idle discharge-jet. When the throttle valve is closed, a high vacuum is created in the manifold of the engine and fuel is thereby lifted up the idle tube and through the idle discharge-jet. Air for atomization is

drawn from below the throttle valve through the idle air-bleed passage under the control of an idle adjustment needle (not shown in Fig. 278).

At speeds from 12 to 20 miles per hour the fuel is delivered into the path of the air stream through both the idle jet and the main discharge-jet. Above 20 miles per hour practically all the gasoline is delivered through the main discharge-jet. The main discharge-jet has its outlet in the center of the smallest section of the Venturi tube; this location insures high air velocity at the tip of the nozzle and effective atomization of the fuel. Atomization of the fuel is aided by bleeding air into

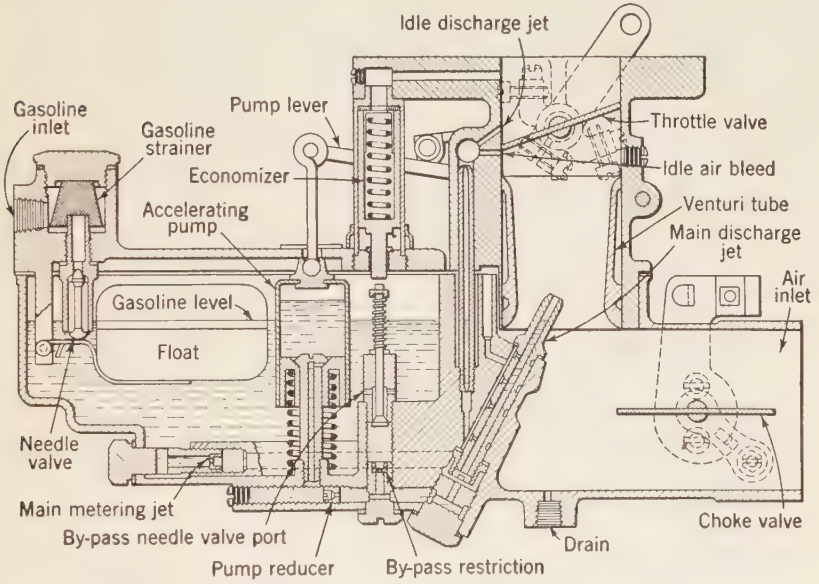


FIG. 278.—Stromberg model "U" carburetor with accelerating pump and vacuum economizer.

the central passage of the main discharge-jet through a series of small holes located below the fuel level.

The economizer is used to enrich the mixture necessary for maximum power and high speed. The movement of the economizer piston is controlled by the suction in the manifold connection of the carburetor. When the throttle is partially open, the high vacuum in the manifold holds this piston in the position shown in Fig. 278. When the throttle is opened, the suction in the manifold is reduced and the economizer piston is forced down by the spring within the hollow piston. This downward movement opens the by-pass needle valve, allowing an additional amount of fuel to flow through the by-pass restriction to the main discharge-jet. When the load on the engine decreases, the

increased vacuum in the cylinder overcomes the force of the spring and the by-pass needle valve is closed, thus reducing the fuel supply to the main discharge-jet.

In order to retain satisfactory fuel economy at normal speeds without sacrificing acceleration and flexibility, it is necessary to momentarily supply an extra amount of fuel when the throttle is suddenly opened. This additional fuel is supplied by the accelerating pump; see Fig. 278. The shaft of the throttle valve is connected (by the pump lever) to the accelerating pump cylinder which contains an auxiliary supply of fuel. When the throttle valve is suddenly opened and the pump cylinder depressed, the piston is forced down by the entrapped gasoline and a port in the upper part of the piston guide is uncovered. The fuel is forced through this passage and the pump reducer passage to the main discharge-jet. While the throttle is held in this position the piston slowly rises under the force of its spring, thus delivering a further accelerating charge to the main discharge-jet. During fixed throttle position, the pump piston is seated against the conical (upper) end of the piston guide, thus closing the accelerating gasoline passage.

The choke valve must be fully closed for starting and partly closed during the period of warming up the engine. Attached to the choke valve shaft is a cam which is used to lift an auxiliary control lever (not shown in Fig. 278) and an auxiliary needle valve from its seat; the latter admits an extra amount of fuel into the air stream at the throat of the Venturi tube when the choke valve is closed or partly closed. This additional fuel enriches the mixture and makes it possible to start and operate a cold engine more easily.

Some carburetors are provided with an auxiliary air valve so arranged that at high engine speeds the valve will open and admit additional air to dilute the mixture. For low speeds, the mixture which gives good results is 12 to 13 parts of air to 1 part of gasoline by weight. At high engine speeds, 15 to 17 parts of air to 1 part of gasoline are usually required for maximum thermal efficiency.

300. The Automobile Engine. The automobile engine, or motor as it is frequently called, is one of the most highly perfected types of internal-combustion engines. The engine used almost exclusively for automobiles is of the multicylinder type operating on the Otto four-stroke cycle. The peculiar demands made by the public have made necessary the building of engines that will withstand considerable abuse. Lubrication, carburetion, cooling, ignition, etc., must be simple and unfailing for long periods of time without any attention.

Figures 279 and 280 show the essential parts of the eight-cylinder engine used in the Packard automobile. The cylinders are cast en bloc (all the cylinders in a single casting) and bolted to an aluminum base

(crankcase). The engine has a detachable cylinder head, the advantages of which are apparent. The poppet valves (an inlet and an

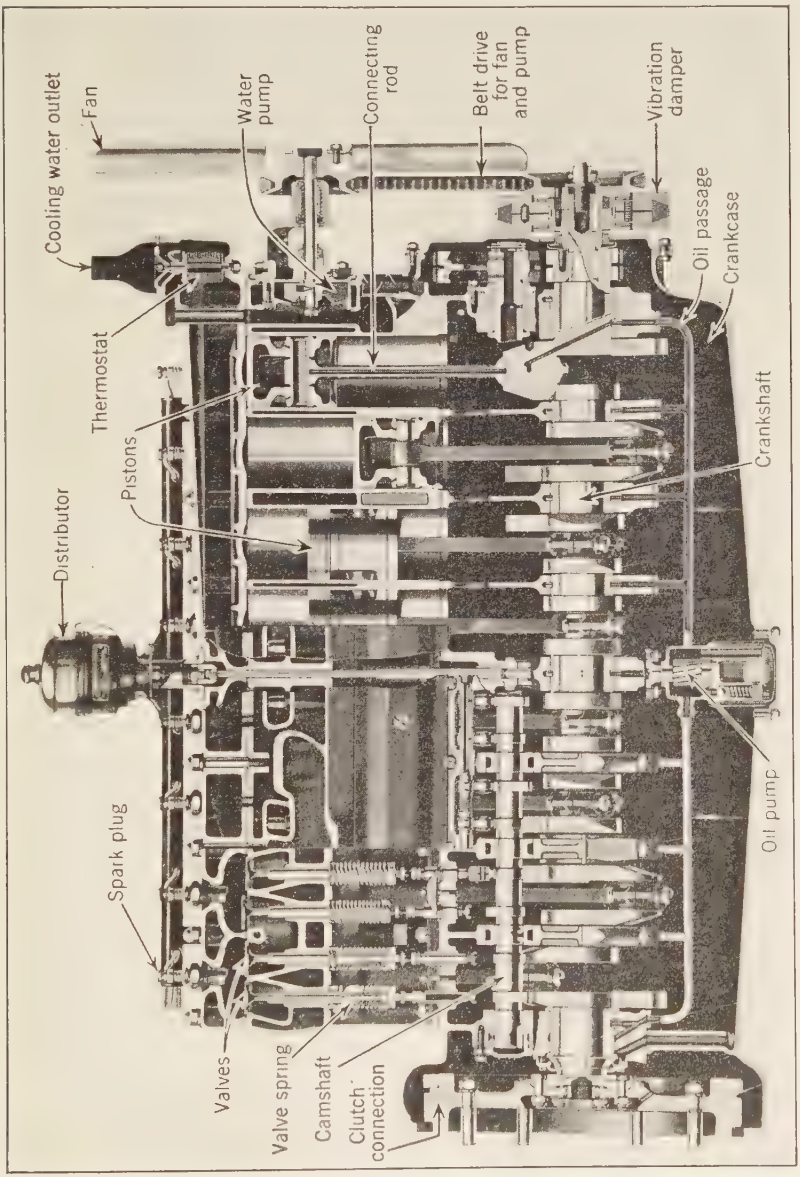


FIG. 279.—Side sectional view, Packard eight-cylinder automobile engine.

exhaust valve for each cylinder) are opened by rocking levers, operated by cams, and are closed by springs. The camshaft is designed to open

and close all the valves at the proper time, so as to give the proper sequence of events in the eight cylinders.

When the piston of one of the cylinders moves downward on the suction stroke the inlet valve is open, and the mixture of gasoline and air is drawn into the cylinder from the carburetor. The piston rises and compresses the mixture, both valves being closed. Compression serves to mix the gasoline vapor more evenly with the air; the contour of the inside of the cylinder head, Fig. 280, also aids this mixing. As the pressure of the mixture increases during compression, the temperature also rises and good ignition is made easier. At the beginning of the next downward stroke the explosive mixture is ignited by an electric spark from a spark plug installed in the top of the cylinder. The combustion resulting from this ignition increases the pressure from about 110 to 300 lb. per sq. in. On the explosion or expansion stroke the piston is forced downward and by means of the connecting rod the driving force is transmitted to the crankshaft of the engine. As the engine has eight cylinders, the torque on the crankshaft is more uniform—there being four power impulses every revolution—than if only one cylinder were employed. On the next upward stroke of the piston the exhaust valve is open and the products of combustion are exhausted to the atmosphere; and the cycle of events, intake, compression, expansion, and exhaust, is repeated.

In order to make the explosions in an engine cylinder as effective as possible, the ignition of the compressed mixture should occur a trifle before the piston reaches the top dead center. Ignition should take place, therefore, just before the end of the compression stroke. The faster the engine is running the earlier in the stroke the ignition should occur, so as to insure complete combustion. That is, the "spark" should be further advanced (occur earlier) at high than at low speeds.

In an automobile engine the intake pipe between the carburetor and the engine is made with as many branches as there are cylinders. These branches of the intake pipe are called the intake manifold. The branches of the manifold going to the intake valves are generally cast in the sides of the cylinders or surrounded by the exhaust manifold. The advantage of this construction is that the manifold can be kept warm by the heat of the engine or exhaust, thus insuring better vaporization of the gasoline.

Cooling of the automobile engine is generally effected by a fan and water jackets. Water for cylinder cooling is circulated through the water jackets and radiator by means of a small rotary pump, located at the rear end of the fan shaft, Fig. 279. The cooling-water circulation may be controlled by a thermostatic valve, Fig. 279. When the engine is cold, there is no circulation to the radiator (top connection). When

the water in the jackets becomes heated the valve opens, and the circulation is through the radiator.

A gear pump, as illustrated in Figs. 279 and 280, is located at the low-

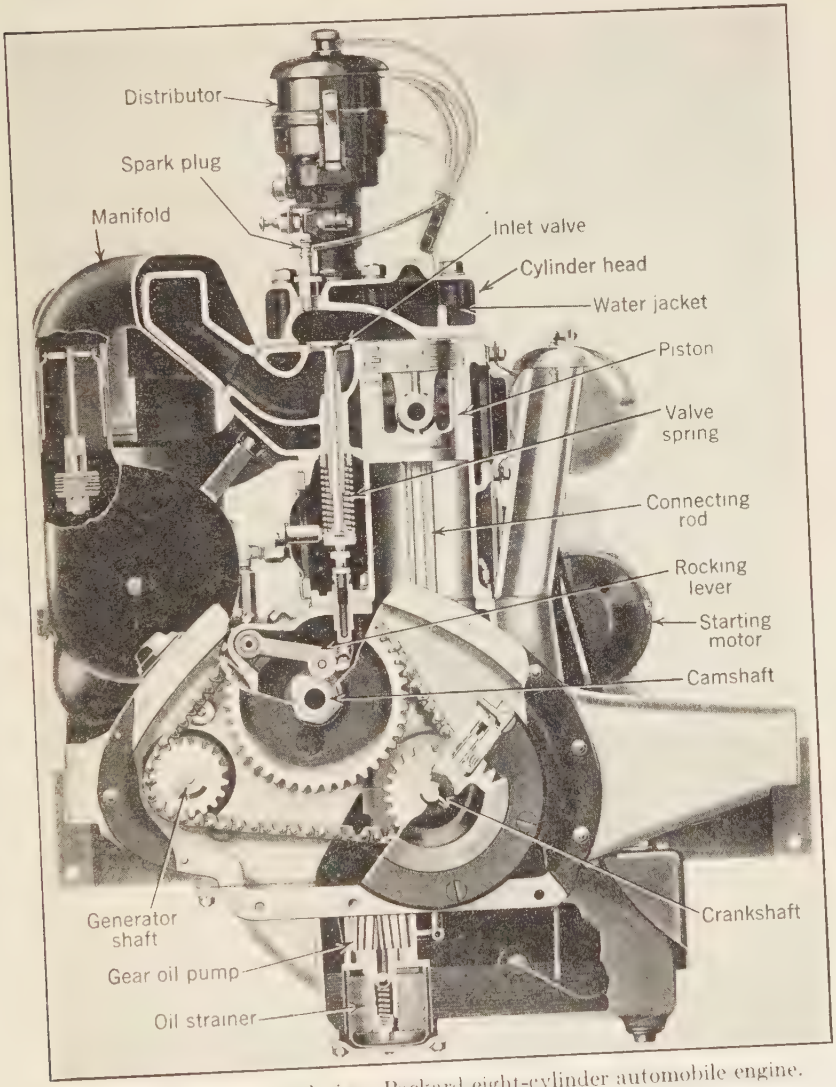


FIG. 280.—End sectional view, Packard eight-cylinder automobile engine.

est point of the crankcase and forces lubricating oil from the reservoir through a system of branch pipes to all parts of the engine and auxiliaries. The pump is operated by a shaft driven through a spiral gear on the camshaft and being submerged in the oil supply requires no

priming. Before entering the pump, the oil is strained through a cylindrical screen which surrounds the entire pump. This strainer may be cleaned by removing a small plate on the bottom of the crankcase.

301. Oil Engines.—An oil engine may be defined as an internal-combustion engine that uses oil for fuel, the oil being sprayed into the cylinder or combustion chamber during or at the end of the compression stroke. Oil engines are of three distinct types: the low-pressure, moderate-pressure, and the Diesel (high-pressure).

The low-pressure type is so named because the compression is only 60 to 140 lb. per sq. in. Fuel is generally injected against hot plates, into a hot-bulb, or into an uncooled vaporizer chamber during the compression stroke. When the atomized fuel comes in contact with the hot surface, vaporization occurs; and as the oil is usually kept out of the cylinder, only the expanding gases come in contact with the piston and cylinder walls. Initial vaporization is often produced by artificially heating the chamber, and after starting the engine the surfaces retain enough heat from explosion to explosion to be continuously effective as a vaporizer. These engines operate on the Otto cycle, and the pressures resulting from the explosion are not high, seldom exceeding 250 lb. per sq. in.

Moderate-pressure oil engines employ compression pressures of 200 to 350 lb. per sq. in. These engines may be designed to operate on cycles very closely approximating the Otto or Diesel, or a combination of the two cycles. Ignition in this type of engine is usually secured by methods similar to those employed in low-pressure oil engines. Injection of the fuel cannot occur during the compression stroke as in the low-pressure type, on account of probable premature ignition. The temperature corresponding to 250 lb. per sq. in. pressure is sufficient to ignite most fuel oils, but this temperature is not high enough to insure perfect ignition at every stroke. In order to avoid occasional pre-ignition, the fuel oil in moderate-pressure oil engines is injected at the end of the compression stroke.

Figure 281 illustrates a typical moderate-pressure oil engine, which operates on the four-stroke Otto cycle having direct injection of the fuel oil. On the suction (outward) stroke of the piston the intake valve is open and air is drawn into the cylinder. Several degrees past the outer dead center the intake valve is closed, and during the remainder of the inward stroke the piston compresses the air into the combustion chamber. Before the inner dead center is reached the fuel pump plunger is actuated by a cam, and two sprays of oil are injected into the combustion chamber. These sprays come together near the center of the combustion space, and the force of their impact materially aids atomization and vaporization. Ignition takes place automatically from the heat

of compression of the charge of air. The fuel injection is so timed that combustion at constant volume increases the cylinder pressure, which is maintained until about 20 deg. after admission. Expansion then begins forcing the piston toward the right; this is the working stroke from which the engine develops power. The piston passes outer dead center and during the return stroke forces the burned gases out through the exhaust-valve opening. The exhaust valve remains open until after the piston has just passed the inner dead center. The suction stroke of the next cycle begins, and the events are repeated. The flywheels (considerably heavier than for steam engines) absorb sufficient energy during the work-

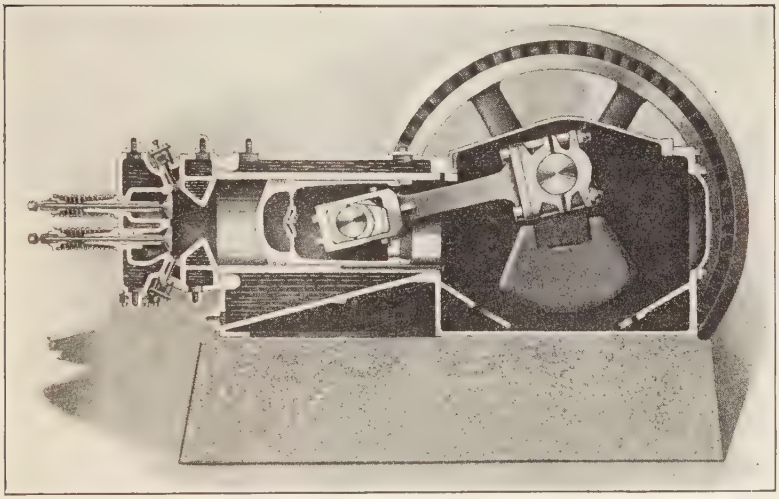


FIG. 281.—Ingersoll-Rand type "PO" oil engine.

ing stroke to maintain a nearly uniform rotary speed of the crankshaft during the suction, compression, and exhaust strokes.

302. The Diesel Engine.—The Diesel engine has been developed, both as a four-stroke and as a two-stroke cycle engine, exclusively as an oil engine, and operates on a cycle, Fig. 274, which approximates the ideal Diesel cycle discussed in Art. 296. Of all types of commercial internal-combustion engines, Diesel engines generally give the best performance, and are used for stationary, marine, and locomotive purposes, especially in the larger sizes. The charge of air is compressed to a high pressure (about 500 lb. per sq. in.), and to a temperature of about 1000 deg. F. At the end of compression, the temperature of the air is sufficiently high to ignite the fuel when injected into the cylinder in the form of a finely divided spray. Combustion of this fuel results at constant pressure; there is no explosion.

When *air injection* of the fuel is employed the pressure of the air for the blast must be considerably higher than the highest cylinder compression pressure. The fuel is injected with a blast of air at a high velocity, and is thus atomized and distributed through the main mass of compressed air in the cylinder. The injection pressure varies from 700 lb. per sq. in. for light loads to 1400 lb. for maximum loads. The auxiliary air is generally compressed in a two- or three-stage compressor driven directly by the engine; the compressor uses 5 to 15 per cent of the power developed.

The elements of a Diesel engine power plant using air injection of the fuel are shown in Fig. 282. The engine illustrated is of the horizontal

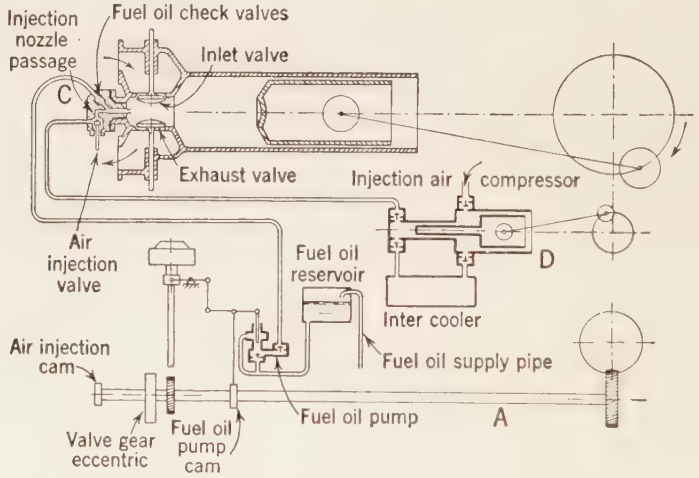


Fig. 282.—Diagrammatic arrangement of a Diesel engine plant.

type operating on the four-stroke cycle. To the crankshaft is geared a lay-shaft A which revolves at one-half the engine speed. On this lay-shaft are mounted the cams used to actuate the admission, exhaust, and fuel-injection valves. The fuel pump is also driven by lay-shaft A, the amount of fuel charge being under control of the governor. The fuel is deposited in the fuel valve C, out of which it is forced by the air charge at the proper instant. The air blast is supplied by the compressor D which is driven by a crank on the end of the engine shaft. In this diagram the air line is not supplied with an air receiver, and the air is delivered directly from the compressor to the fuel valve. In addition to the valves shown, one or more cylinders of an engine are provided with "starting valves," which are used to supply compressed air to these cylinders for starting the engine.

The power used by the injection-air compressor may be eliminated, the mechanism of the engine simplified, and the first cost decreased by using *solid injection* of the fuel (also called airless injection or mechanical injection). Oil is forced into the cylinder at the proper time, through specially designed nozzles that produce the necessary fineness of spray, by one or more pumps under a pressure of 3000 lb. per sq. in. or more.

To reduce the size, weight, and cost for a given power, Diesel engines are built to operate on the two-stroke cycle. A number of recent installations of this kind have shown fuel economies (pounds of oil used per brake horsepower per hour) approximately the same as with four-stroke engines, especially in the larger sizes. Figure 283 shows a sectional view of a two-cycle solid-injection Diesel engine built in sizes from 60 to 540 brake hp. The " crank " end of the cylinder acts as a scavenging pump; this design requires the use of a piston rod and crosshead. The piston and piston rod are made hollow, and oil under pressure is used for cooling the piston. The down stroke of the piston compresses the air on the under side of the piston and the air is delivered to receiver R, passing through " Feather " valves. An inclined port is employed to give the proper direction to the scavenging air as it blows into the cylinder at a pressure of about 3 lb. per sq. in. The exhaust ports E are uncovered by the piston near the end of the expansion stroke, and exhaust begins. The piston next uncovers the inclined air port, and, when the pressure within the cylinder falls below that of the pressure in R, air enters to scavenge the cylinder and to provide a charge of air for the compression stroke. On the upward stroke of the piston the charge of air is compressed in the upper end of the cylinder to a pressure of nearly 500 lb. per sq. in. and air is drawn into the lower end of the cylinder.

Two-stage or divided combustion is used in this engine, Fig. 283. The fuel pump injects oil in an atomized form into the pre-combustion chamber, top of cylinder Fig. 283, just before the end of compression. The fuel is partially burned in this small space during injection, then is blown out into the combustion space of the power cylinder to be completely consumed in the main air charge at nearly constant pressure. The movement of the fuel from the first stage to the second stage is determined and controlled mainly by the downward movement of the piston, which causes a differential pressure between the small spherical chamber and the top of the cylinder. However, the spraying of the fuel into the cylinder is started and assisted by the pre-combustion that takes place during injection. Combustion in this type of engine is fixed by the size and shape of the injection (pre-combustion) chamber, the outlet nozzle arrangement, and the spray. Two-stage combustion has

the advantage of limiting the rate of burning of the oil during injection, so that accurate timing and careful injection control are not necessary. In the timing of direct mechanical injection, wherein the fuel meets the whole air charge at once, the combustion control required is represented by a few degrees of crank angle.

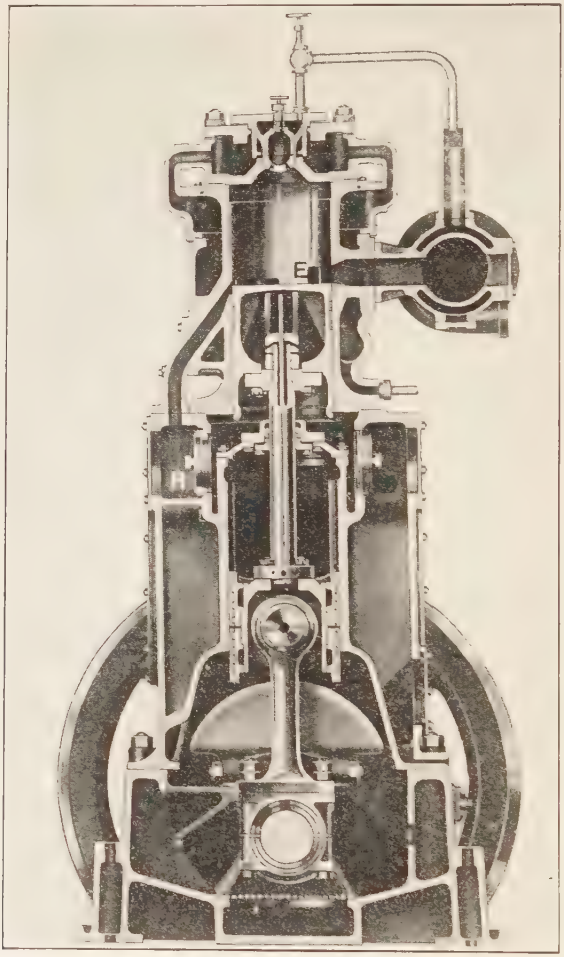


FIG. 283.—Section of a Worthington two-cycle solid-injection Diesel engine.

Diesel engines may be built double-acting in order to further reduce the size, weight, and cost for a given power. The usual arrangement employs piston rods and crossheads, the pistons and rods being made hollow in order that oil or water under pressure may be used for cooling. The high temperatures to which the pistons in Diesel engines are sub-

jected limits their diameter to about 20 in. unless some means of cooling is employed. Engines have been built to develop 2500 hp. per cylinder, with water-cooled pistons as large as 35 in. in diameter, in the double-acting two-stroke Diesel-cycle types.

303. Governing.—The governing of internal-combustion engines does not differ materially from the governing of steam engines. The purpose of engine governors is to maintain approximately constant speed of rotation and to obtain the maximum thermal efficiency at all loads.

Automobile and marine engines are usually hand-governed. There are three general types of internal-combustion-engine governors: (1) hit-and-miss governors, (2) quantity governors, and (3) quality governors.

With the *hit-and-miss* type of governor there is a working stroke for every cycle under conditions of maximum load. At lighter loads, when the speed increases, the governor mechanism acts to prevent admission of a charge of fuel occasionally and there is no explosion, causing the engine to "miss." When a "miss" occurs not only does the inlet valve remain closed, but also the exhaust valve is generally held open, so that, during the strokes corresponding to the ordinary cycle, the piston pumps exhaust gas into and out of the exhaust pipe with practically no loss of energy. The loss of a power stroke decreases the speed of the engine; the governor mechanism opens the inlet valve, an explosion or a "hit" occurs, and the engine receives a power impulse. With this method of governing, the engine either operates under conditions of maximum efficiency or does not fire at all. Hit-and-miss governing gives better economy at light loads than the other methods, but does not give close speed regulation. Considerable intervals of time may intervene between power strokes of the engine, and thus a heavy flywheel is required to avoid considerable variation in the speed. Hit-and-miss governing may be used for engines which do not require close speed regulation and with the smaller sizes, less than 40 brake hp.

Quantity governing may be accomplished by varying the amount of mixture entering the cylinder while the proportions of the mixture and the number of cycles are constant. This result may be obtained in one of two ways: (1) cutting off the charge before the piston reaches the end of the suction stroke; (2) throttling, except at full load, the charge during the suction stroke to reduce the amount of mixture entering the cylinder. The compression varies with the charge taken into the cylinder; reducing the compression lowers the efficiency, hence quantity governing is not so efficient for a large variation of load as the hit-and-miss type.

The *quality* governor varies the proportion of fuel to air, but the number of cycles and the quantity of charge per cycle remain the same.

At full load a rich mixture is used, and at low loads a lean mixture. Mechanically, this method is quite simple, but low thermal efficiencies result at light loads. Difficulty may be encountered in securing satisfactory ignition with very rich or very lean mixtures. Such mixtures burn very slowly, and combustion continues in some cases throughout the entire expansion stroke.

A number of combinations of the foregoing governors have been used. Thus quality governing may be used for the high and medium loads, and the hit-and-miss governor for the light loads. Close regulation may be obtained by employing quality governing at high loads and changing to quantity governing at low loads. These combinations are desirable, but they tend to complicate the valve gear and require accurate adjustments.

The automobile engine is hand-governed by a quantity (throttle) control of the charge entering the cylinders, the proportion of gasoline to air remaining the same for a given carburetor adjustment; see page 444.

Oil engines use quality governing, the same amount of air being drawn into the cylinder during each suction stroke, with the governor controlling the amount of fuel injected per cycle. The amount of fuel forced into the cylinder may be governed by varying the length of the stroke of the fuel pump or by by-passing the fuel so that more or less is injected into the cylinder.

304. Starting and Lubrication.—A steam engine is started by turning the steam under pressure into the cylinder. An internal-combustion engine cannot be started in this way because there is no pressure to be turned into the cylinder, unless compressed air is used. Large engines are started with compressed air in the same way that steam is used in a steam engine. Small engines are set in motion by an electric motor, by compressed air, or by hand, the starting operation continuing until the explosions of the fuel begin in the cylinder.

Lubrication is a matter of great importance in the operation of internal-combustion engines, because the rubbing surfaces are generally dry, both the temperature and the pressure are high, and the speed is high. Oil is usually forced by pump pressure direct to the crankshaft bearings and, by means of drilled holes in the crank web, to the crank pins, thence through oil pipes attached to the connecting rods or through hollow connecting rods (see Fig. 279) to the piston pins. The pistons and cylinders may be lubricated by oil thrown from the crank ends of the connecting rods.

305. Performance. –The horsepower rating of an internal-combustion engine required for driving a machine is greater than that of a steam engine or turbine for driving the same machine. The gas engine as rated has generally no overload capacity, whereas a steam engine or turbine

operates successfully at a 25 per cent overload. Internal-combustion engines are commonly rated on the maximum brake horsepower output; steam engines are usually rated on the indicated horsepower at which the best steam economy is obtained.

The indicated horsepower of an internal-combustion engine is determined in the same way as for the steam engine. The mean effective pressure is determined from indicator diagrams, which show the variation of the pressure in the engine cylinder at all points of the cycle. The remaining data required for the calculation of the indicated horsepower are the number of explosions per minute and the dimensions of the engine cylinder.

$$\text{i.hp.} = \frac{p_m l A N'}{33{,}000}, \tag{184}$$

where p_m = mean effective pressure, lb. per sq. in.
l = length of stroke, ft.
A = cross-sectional area of cylinder, sq. in.
N' = number of *explosions* per minute.

When an explosion occurs during every cycle: (1) for a single-acting four-stroke engine, the number of explosions per minute will equal one-half the revolutions per minute of the crankshaft; (2) for a single-acting two-stroke engine the number of explosions per minute will equal the revolutions per minute of the crankshaft; (3) for a double-acting two-cycle engine the number of explosions per minute will be twice the revolutions per minute. As in steam engines, the total indicated horsepower of a multicylinder internal-combustion engine is the sum of the horsepowers developed in the individual cylinders.

The manufacturer invariably works on a brake horsepower basis, on account of the difficulties encountered in accurately measuring the indicated horsepower. The accuracy of the indicator diagram may be effected by the high speed, high temperature, and high pressure of the engine. Special indicators, such as optical and diaphragm types, are generally used to obtain indicator diagrams, when the engine speed exceeds 500 r.p.m., and for greater accuracy at lower speeds. The brake horsepower of a gas engine is determined in exactly the same way as for the steam engine.

$$\text{b.hp.} = \frac{2\pi r w N}{33{,}000}, \tag{185}$$

where r = length of the brake arm, ft.
w = the net weight on the brake, lb.
N = the number of *revolutions* per minute.

The *mechanical efficiency* of an engine is the ratio of the brake horsepower output to the indicated horsepower developed.

$$e_m = \frac{\text{b.hp.}}{\text{i.hp.}}. \tag{186}$$

The difference between the indicated and brake horsepower is known as the mechanical loss or friction horsepower and includes the negative loop (pumping loss) of the indicator diagram.

The *thermal efficiency* based on the brake horsepower shows what fraction, or percentage, of the heat supplied was actually delivered by the engine.

$$e_t = \frac{2545}{W_f \times H_f}, \tag{187}$$

where 2545 = B.t.u. equivalent of one horsepower per hour.
W_f = weight of fuel supplied, lb. per b.hp. per hr.
H_f = high heat value, B.t.u. per lb. of fuel supplied.

The heat supplied to an internal-combustion engine in the fuel may be accounted for in a "heat balance." This shows the heat converted into work (brake horsepower); and the heat lost: in overcoming friction, exhaust gases, water jacket, lubricating oil, radiation, leakage, etc.

The fuel consumption, measured in cubic feet of gas per brake horsepower per hour, or pounds of oil per brake horsepower per hour, varies considerably with the different types and sizes of internal-combustion engines. The fuel used by engines of a given type decreases with increase of cylinder diameter: the decrease may be considerable as the size progresses. The single-cylinder engine is invariably the most economical for a given size. However, practical difficulties limit the size to which cylinders may be built without complicating the design, so that multicylinder engines are used for large sizes.

The fuel economy and the thermal efficiency for a given engine will vary with the load, Fig. 284; these remain nearly constant from three-quarter load to full load. Other types of internal-combustion engines give results somewhat similar to those illustrated in Fig. 284.

Modern multicylinder gas engines are guaranteed to produce a brake horsepower hour for each 9000 to 10,000 B.t.u. supplied in the gas to the engine, corresponding to a thermal efficiency of 28.3 to 25.5 per cent. The modern automobile engine has a fuel economy of 0.60 to 0.80 lb. gasoline per b.hp. per hr. when operating at or near the rated load and speed, with an air-fuel ratio of about 16 to 1 (by weight). Low-pressure (60- to 140-lb. compression) oil engines consume approximately 0.55 to 0.80 lb. of fuel oil per b.hp. per hr., whereas for moderate-pressure

(200- to 300-lb. compression) oil engines average values are 0.45 to 0.70 lb. of fuel oil per b.hp. per hr. for operation at one-half to full load.

Some typical performance curves of a modern Diesel engine, operating under various loads, are shown in Fig. 284. From one-half to full load the fuel consumption varies from 0.46 to 0.42 lb. per b.hp. per hr. with a thermal efficiency of 29.4 to 32.2 per cent. Some Diesel engine tests have shown fuel economies of less than 0.40 lb. per b.hp. per hr. with a resultant thermal efficiency of 35 to 40 per cent. These figures give some idea as to the utility of internal-combustion engines and the future

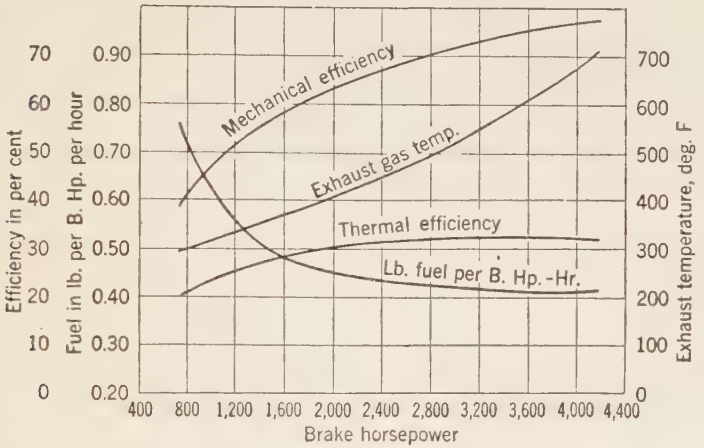


FIG. 284.—Performance curves of a 3750-b.hp. Diesel engine; 29 in. by 44 in., six-cylinder, single-acting, vertical, two-cycle, 125 r.p.m. fuel-oil (18,740 B.t.u. per lb.), air injection.

possibilities for this type of prime mover, when considered from a standpoint of low fuel consumption and minimum operating cost.

PROBLEMS

1. Determine the air standard thermal efficiency for an internal-combustion engine operating on the Otto cycle with a volumetric clearance of 30 per cent: (*a*) when $k = 1.4$, (*b*) when $k = 1.3$?

2. Calculate the ideal thermal efficiency of a gas engine operating on the Otto cycle with a pressure at the beginning of compression of 14 lb. per sq. in. abs.: (*a*) when the compression pressure is 70 lb. per sq. in. abs., and (*b*) when the compression pressure is 140 lb. per sq. in. abs. $k = 1.4$.

3. Find the clearance in per cent of the stroke for an Otto-cycle engine that will give an air standard thermal efficiency of 48 per cent. $k = 1.4$.

4. The pressure at the beginning of compression in an Otto-cycle engine was 14 lb. per sq. in. abs. (*a*) Find the pressure required at the end of compression for an air standard thermal efficiency of 52 per cent. (*b*) What clearance was required in per cent of the stroke?

5. Find the clearance in per cent of the stroke for an Otto engine that will give 120 lb. per sq. in. abs. pressure at the end of adiabatic compression when the pressure at the beginning of compression is 14.5 lb. per sq. in. abs.

6. A 10-in. by 18-in. Otto engine has a 25 per cent clearance. At the beginning of the compression stroke the pressure was 14 lb. per sq. in. abs., and the temperature was 180 deg. F. (*a*) What was the air standard thermal efficiency? (*b*) Find the pressure and temperature at the end of compression.

7. Determine the air standard thermal efficiency of an internal-combustion engine operating on the Otto cycle for clearance volumes of 2, 5, 10, 15, 20, 30, and 40 per cent. $k = 1.4$. Plot a curve showing the variation of efficiency with clearance.

8. The pressure at the beginning of compression in an Otto cycle was 13.7 lb. abs. and the pressure at the end of compression was 100 lb. abs. If the clearance was 28 per cent, find the numerical value of the exponent n of the compression line.

9. By a method similar to that used in finding the theoretical m.e.p. of a steam engine cylinder, find the m.e.p. of an internal-combustion engine operating on the ideal Otto cycle. The pressure at the beginning of compression was 14.5 lb. per sq. in. abs., pressure at the end of compression 87 lb. per sq. in. abs., and the pressure at the end of combustion 250 lb. per sq. in. abs. Use $k = 1.4$.

10. Find the air standard thermal efficiency of a Diesel engine having 6 per cent clearance when the fuel is injected for 8 per cent of the stroke.

11. The pressure in the cylinder of a Diesel engine at the beginning of compression was 14 lb. per sq. in. abs. and at the end of compression 560 lb. per sq. in. abs. (*a*) Find the volumetric clearance in per cent if the compression was according to $PV^{1.35} = C$. (*b*) What was the temperature at the end of compression, if the temperature at the beginning of compression was 180 deg. F.?

12. Assume that it is desired to obtain a temperature of 1000 deg. F. at the end of compression in a Diesel cycle. The initial conditions are 13.5 lb. per sq. in. abs. and 120 deg. F. Find the volumetric clearance in per cent and the pressure at the end of compression: (*a*) when compression is according to $PV^{1.4} = C$; (*b*) when $PV^{1.35} = C$; and (*c*) when $PV^{1.3} = C$.

13. Determine the air standard thermal efficiency of a Diesel engine with 5 per cent clearance when the fuel is injected for 2, 5, 10, 15, 20, and 25 per cent of the stroke. $k = 1.4$. Plot a curve showing the variation of efficiency with fuel cut-off.

14. Calculate the air standard thermal efficiency for Diesel engines having volumetric clearances of 2, 4, 6, 10, 12, and 15 per cent, with fuel injection for 10 per cent of the stroke in each case. $k = 1.4$.

15. A four-cycle, single-acting, single-cylinder, 12-in. by 18-in. gas engine was operated at 200 r.p.m. and made 98 explosions per minute. The gross weight on the brake was 260 lb., the tare 40 lb., and the length of the brake arm 60 in. The area of the indicator card was 1.44 sq. in. and the length 3 in. The scale of the indicator spring used was 200 lb. Find: (*a*) the indicated horsepower developed; (*b*) the brake horsepower delivered; (*c*) the friction horsepower; (*d*) the mechanical efficiency; and (*e*) the thermal efficiency, if 400 cu. ft. of natural gas were supplied per hour, high heat value of gas was 1050 B.t.u. per cu. ft., based on b.hp.

16. A four-cycle, single-acting, single-cylinder, 12-in. by 21-in. gas engine rated at 40 b.hp. had a gas consumption of 12,500 B.t.u. per b.hp-hr. Find: (*a*) the thermal efficiency; (*b*) the cost of operation per hour at rated load, if the gas had a high heat value of 500 B.t.u. per cu. ft. and cost $0.75 per 1000 cu. ft.; and (*c*) how the cost of gas compares with the cost of operating an electric motor (to deliver 40 b.hp.) that is 85 per cent efficient, when power costs 3 cents per kw-hr.

17. A four-cycle, single-acting, single-cylinder, $13\frac{1}{2}$-in. by 17-in. gas engine rated at 60 b.hp. required 24 cu. ft. of gas per b.hp-hr. at 200 r.p.m.; high heat value of gas supplied was 460 B.t.u. per cu. ft. Find: (*a*) the suction displacement in cubic feet per brake horsepower per minute; (*b*) the m.e.p. based on rated brake horsepower; (*c*) the B.t.u. supplied to the cylinder per cubic foot of suction displacement; and (*d*) the thermal efficiency in per cent.

18. A two-cycle, single-acting, six-cylinder, 29-in. by 44-in. Diesel engine was guaranteed to deliver 3750 b.hp. at 125 r.p.m. with a fuel consumption of 0.45 lb. per b.hp-hr.; high heat value of fuel oil was 18,740 B.t.u. per lb. Find: (*a*) the suction displacement in cubic feet per brake horsepower per minute.; (*b*) the m.e.p. based on rated brake horsepower; (*c*) the B.t.u. supplied to the cylinder per cubic foot of suction displacement; and (*d*) the thermal efficiency in per cent.

19. A marine Diesel engine that delivered 3330 b.hp. had a mechanical efficiency of 80.3 per cent. The thermal efficiency based on the brake horsepower was 33.2 per cent, and the heat lost to the water jacket was 22 per cent. How many gallons of cooling water were circulated per minute, if the water entered the jacket at a temperature of 65 deg. F. and left at 105 deg. F.?

20. The following data were obtained during a test of a four-cycle, single-acting, single-cylinder, 12-in. by 18-in. oil engine: duration of test, 45 min.; oil used, 12 lb.; total revolutions, 9090; jacket water, 1122 lb.; rise of temperature of jacket water, 50 deg. F.; m.e.p. in cylinder, 94 lb. per sq. in.; torque (*rw*) due to brake load, 840 ft-lb.; high heat value of fuel oil, 18,800 B.t.u. per lb. Find: (*a*) the indicated and brake horsepowers, also the mechanical efficiency; (*b*) the oil used, pounds per indicated horsepower-hour and pounds per brake horsepower-hour; (*c*) the per cent of total heat supplied that was converted into brake horsepower; (*d*) the per cent of total heat supplied that was absorbed by the water jacket; and (*e*) the per cent of heat lost by friction, exhaust gases, radiation, etc.

APPENDIX

ADDITIONAL TABLES

TABLE XXI

DENSITY OF WATER AT VARIOUS TEMPERATURES AT SATURATION PRESSURE
(From A.S.M.E. Steam Tables)

Temp., Deg. F.	Weight, Lb. per Cu. Ft.	Temp., Deg. F.	Weight, Lb. per Cu. Ft.	Temp., Deg. F.	Weight, Lb. per Cu. Ft.
20	62.37	190	60.39	420	52.74
30	62.42	200	60.13	440	51.87
40	62.43	210	59.92	460	51.02
50	62.42	220	59.67	480	50.00
60	62.37	230	59.42	500	48.78
70	62.30	240	59.17	520	47.62
80	62.22	250	58.89	540	46.30
90	62.11	260	58.62	560	44.84
100	62.00	270	58.34	580	43.29
110	61.86	280	58.04	600	41.49
120	61.71	290	57.74	620	39.37
130	61.55	300	57.41	640	37.17
140	61.38	320	56.75	660	34.48
150	61.20	340	56.02	680	31.06
160	61.01	360	55.25	690	28.82
170	60.79	380	54.47	700	25.38
180	60.61	400	53.62	706.1	19.16

TABLE XXII

Areas of Circles

Diameter	Area	Diameter	Area	Diameter	Area	Diameter	Area
$\frac{1}{8}$	0.0123	10	78.54	30	706.86	65	3318.3
$\frac{1}{4}$	0.0491	$10\frac{1}{2}$	86.59	31	754.76	66	3421.2
$\frac{3}{8}$	0.1104	11	95.03	32	804.24	67	3525.7
$\frac{1}{2}$	0.1963	$11\frac{1}{2}$	103.86	33	855.30	68	3631.6
$\frac{5}{8}$	0.3068	12	113.09	34	907.92	69	3739.2
$\frac{3}{4}$	0.4418	$12\frac{1}{2}$	122.71	35	962.11	70	3848.4
$\frac{7}{8}$	0.6013	13	132.73	36	1017.8	71	3959.2
1	0.7854	$13\frac{1}{2}$	143.13	37	1075.2	72	4071.5
$1\frac{1}{8}$	0.9940	14	153.93	38	1134.1	73	4185.4
$1\frac{1}{4}$	1.227	$14\frac{1}{2}$	165.13	39	1194.5	74	4300.8
$1\frac{3}{8}$	1.484	15	176.71	40	1256.6	75	4417.8
$1\frac{1}{2}$	1.767	$15\frac{1}{2}$	188.69	41	1320.2	76	4536.4
$1\frac{5}{8}$	2.073	16	201.06	42	1385.4	77	4656.6
$1\frac{3}{4}$	2.405	$16\frac{1}{2}$	213.82	43	1452.2	78	4778.3
$1\frac{7}{8}$	2.761	17	226.98	44	1520.5	79	4901.6
2	3.141	$17\frac{1}{2}$	240.52	45	1590.4	80	5026.5
$2\frac{1}{4}$	3.976	18	254.46	46	1661.9	81	5153.0
$2\frac{1}{2}$	4.908	$18\frac{1}{2}$	268.80	47	1734.9	82	5281.0
$2\frac{3}{4}$	5.939	19	283.52	48	1809.5	83	5410.6
3	7.068	$19\frac{1}{2}$	298.64	49	1885.7	84	5541.7
$3\frac{1}{4}$	8.295	20	314.16	50	1963.5	85	5674.5
$3\frac{1}{2}$	9.621	$20\frac{1}{2}$	330.06	51	2042.8	86	5808.8
$3\frac{3}{4}$	11.044	21	346.36	52	2123.7	87	5944.6
4	12.566	$21\frac{1}{2}$	363.05	53	2206.1	88	6082.1
$4\frac{1}{2}$	15.904	22	380.13	54	2290.2	89	6221.1
5	19.635	$22\frac{1}{2}$	397.60	55	2375.8	90	6361.7
$5\frac{1}{2}$	23.758	23	415.47	56	2463.0	91	6503.9
6	28.274	$23\frac{1}{2}$	433.73	57	2551.7	92	6647.6
$6\frac{1}{2}$	33.183	24	452.39	58	2642.0	93	6792.9
7	38.484	$24\frac{1}{2}$	471.43	59	2733.9	94	6939.8
$7\frac{1}{2}$	44.178	25	490.87	60	2827.4	95	7088.2
8	50.265	26	530.93	61	2922.4	96	7238.2
$8\frac{1}{2}$	56.745	27	572.55	62	3019.0	97	7389.8
9	63.617	28	615.75	63	3117.2	98	7542.9
$9\frac{1}{2}$	70.882	29	660.52	64	3216.9	99	7697.7

TABLE XXIII

Common Logarithms of Numbers

	0	1	2	3	4	5	6	7	8	9
100	0000	0004	0009	0013	0017	0022	0026	0030	0035	0039
101	0043	0048	0052	0056	0060	0065	0069	0073	0077	0082
102	0086	0090	0095	0099	0103	0107	0111	0116	0120	0124
103	0128	0133	0137	0141	0145	0149	0154	0158	0162	0166
104	0170	0175	0179	0183	0187	0191	0195	0199	0204	0208
105	0212	0216	0220	0224	0228	0233	0237	0241	0245	0249
106	0253	0257	0261	0265	0269	0273	0278	0282	0286	0290
107	0294	0298	0302	0306	0310	0314	0318	0322	0326	0330
108	0334	0338	0342	0346	0350	0354	0358	0362	0366	0370
109	0374	0378	0382	0386	0390	0394	0398	0402	0406	0410
110	0414	0418	0422	0426	0430	0434	0438	0441	0445	0449
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981

Note.—The Napierian logarithm of a number equals 2.3026 times common logarithm of that number or

$$\log_e n = 2.3026 \times \log_{10} n$$

TABLE XXIII—*Continued*

COMMON LOGARITHMS OF NUMBERS

	0	1	2	3	4	5	6	7	8	9
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996

NOTE.—The Napierian logarithm of a number equals 2.3026 times common logarithm of that number or

$$\log_e n = 2.3026 \times \log_{10} n$$

TABLE XXIV

Properties of Dry Air and Saturated Air

Absolute Pressure, 29.921 Inches of Mercury

Temperature, Degrees F.	Weight of 1 Cubic Foot of Dry Air, Pound	Volume of 1 Pound of Dry Air, Cubic Feet	Volume of 1 Pound of Dry Air Plus Vapor to Saturate it, Cubic Feet	Weight of Water Necessary to Saturate 1 Cubic Foot of Dry Air, Pound	Weight of Water Necessary to Saturate 1 Pound of Dry Air, Pounds
0	0.08636	11.58	11.59	0.0000674	0.000781
10	0.08453	11.83	11.86	0.0001103	0.001309
20	0.08276	12.08	12.12	0.000177	0.002144
30	0.08107	12.34	12.41	0.000278	0.003444
35	0.08025	12.46	12.54	0.000340	0.004268
40	0.07945	12.59	12.70	0.000410	0.005202
45	0.07866	12.71	12.84	0.000492	0.00632
50	0.07788	12.84	13.00	0.000588	0.00764
55	0.07713	12.97	13.16	0.000699	0.00920
60	0.07640	13.09	13.32	0.000829	0.01105
65	0.07567	13.22	13.50	0.000979	0.01323
70	0.07495	13.34	13.68	0.001153	0.01578
75	0.07424	13.47	13.87	0.001352	0.01877
80	0.07356	13.59	14.08	0.001580	0.02226
85	0.07289	13.72	14.30	0.001841	0.02634
90	0.07222	13.85	14.54	0.002137	0.03109
95	0.07157	13.97	14.79	0.002474	0.03662
100	0.07093	14.10	15.07	0.002855	0.04305
110	0.06968	14.35	15.72	0.003769	0.0593
120	0.06848	14.60	16.50	0.004920	0.0813
130	0.06732	14.85	17.50	0.006356	0.1114
140	0.06620	15.11	18.82	0.008130	0.1532
150	0.06510	15.36	20.57	0.01030	0.2122
160	0.06406	15.61	23.06	0.01294	0.2987
170	0.06304	15.86	26.80	0.01611	0.4324
180	0.06205	16.12	33.00	0.01991	0.6577
190	0.06110	16.37	44.96	0.02441	1.0985
200	0.06018	16.62	77.19	0.02972	2.2953
220	0.05840	17.12			
240	0.05673	17.63			
260	0.05516	18.13			
280	0.05367	18.63			
300	0.05225	19.14			
350	0.04903	20.40			
400	0.04618	21.65			
450	0.04364	22.91			
500	0.04138	24.17			
550	0.03932	25.43			
600	0.03746	26.70			
700	0.03423	29.21			
800	0.03151	31.74			
900	0.02920	34.25			
1000	0.02720	36.76			
1200	0.02392	41.81			

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enthalpy = internal energy + volume × pressure

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